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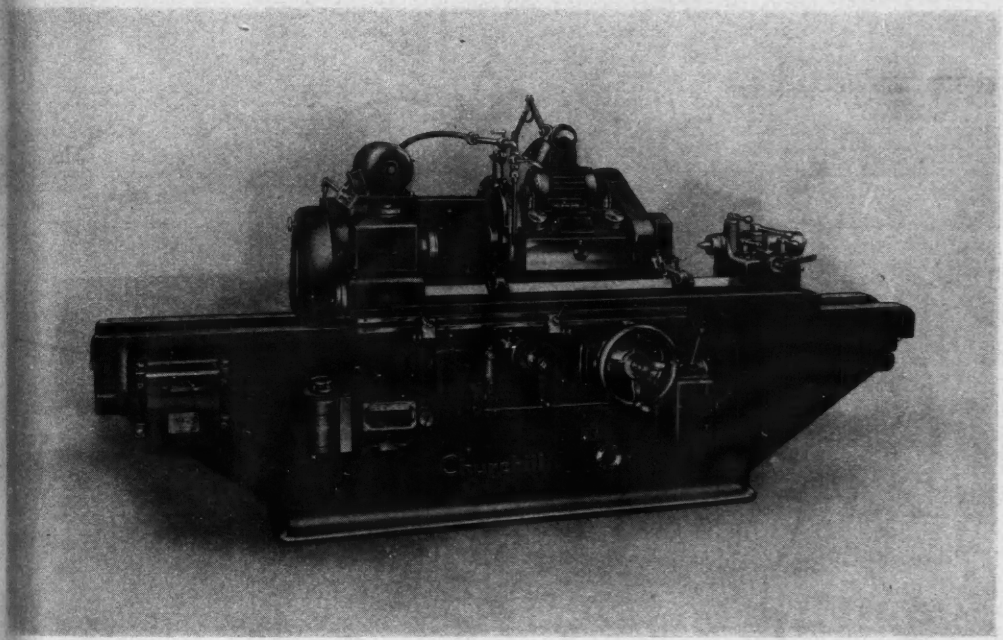

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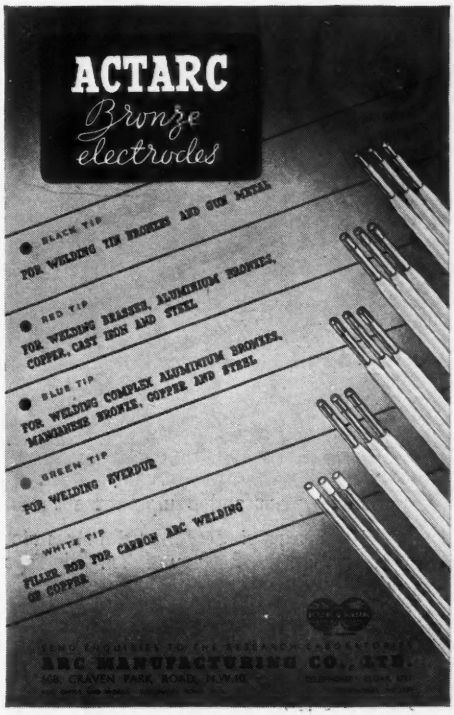
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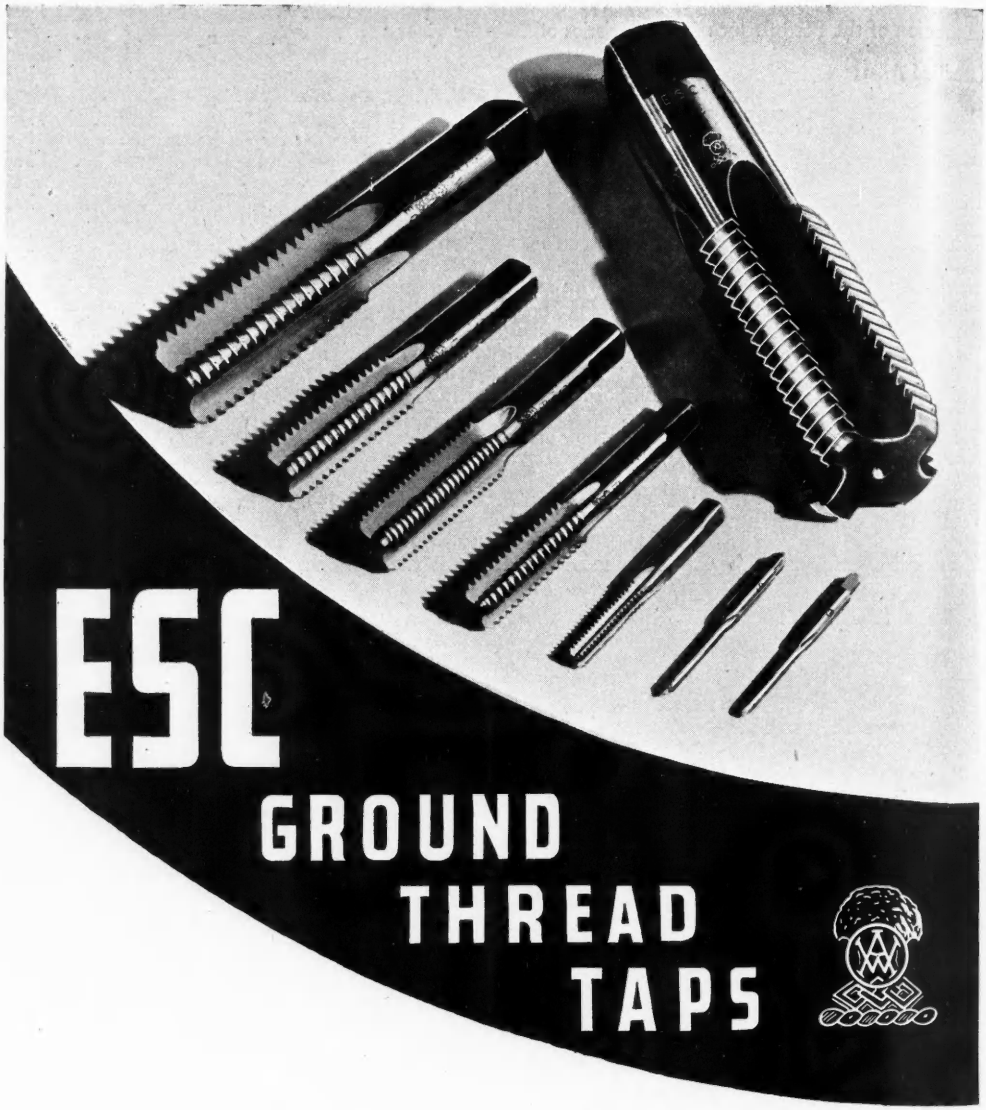

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THE EFFICIENT OPERATION OF SEMI-GAS FURNACES.

By K. GUTHMANN. (From *Stahl und Eisen*, Vol. 63, No. 36, September 9, 1943, pp. 659-663, 2 illustrations.)

ALONG with gas furnaces with their numerous operating advantages, one must still give consideration to the semi-gas furnace. Having regard to the much lower price of solid fuel per calory content, it is found that such furnaces can be most economically and efficiently operated, with a heat utilization of between 60 and 100 kgs. (132·3 and 220·45 lbs). " Normal " coal per ton charge weight, at 420 to 700·103 kcal per ton charged weight. Also hearth performances of the order of 130 to 270 kgs. per sq. metre (26·65 *to* 55·35 *lbs./sq. ft.*) or more can be considered good. It is quite wrong to consider semi-gas furnaces as out of date. In some works it has been found advisable to go back to semi-gas firing from gas firing, while in others, for the same reasons, old semi-gas furnaces, in parts 20 years old, have been retained, on which much development has taken place in the last 10 years.

Fig. 1 shows a semi-gas pusher type furnace for use with suitable secondary air introduced just behind the fire bridge which results in the gases being completely burnt and directed down on the stock so as to produce thorough soaking. At the same time, this method of working results in a cooling layer of air under the arch which protects the refractories, while there is a reducing atmosphere over the stock to prevent undue scaling. The furnace has a step-grate which gives substantially better combustion than the plain level bar grate. Over this is a shute for the coal. Primary air is introduced right and left under the furnaces, and undue clinkering of the coal on the grate is prevented by some such method as dampening the primary air with steam or water sprinkler.

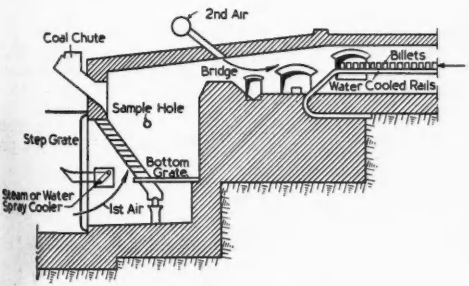


Fig. 1. Semi-gas firing of pusher type furnace.

OPERATING DATA

The following grate area loadings can be considered satisfactory with semi-gas furnaces :—

Without primary or secondary air—75 to 100 kgs. stone coal per sq. m. (15·375 *to* 20·5 *lbs./sq. ft.*) grate area or more.

With primary or secondary air—up to 230 kgs. stone coal per sq. m. (47·15 *lbs./sq. ft.*) grate area, and higher with piece coal.

Combined bar and step grate for Fazing or Mill Furnaces (Fig. 1)—60 to 180 kgs. stone coal per sq. m. (12·3 *to* 36·9 *lbs./sq. ft.*) total grate area and more. 100 to 180 kgs. stone coal per sq. m. (20·25 *to* 36·9 *lbs./sq. ft.*) step grate area and more.

Combined bar and step grate for Annealing furnaces—50 to 120 kgs. per sq. m. (10·25 *to* 24·6 *lbs./sq. ft.*) grate area and more.

For example, with a coal consumption of 1230 kgs. (2711 *lbs.*) of coal per hour and a grate area of 9 sq. metres (96·87 *sq. ft.*) a grate area loading of 137 kgs. per sq. m. (28 *lbs./sq. ft.*) would result.

With " Normal " coal with a lower calorific value of 7,000 kcal per kg. (1 kcal = 4 B.T.U's) the theoretical air requirement is 7·6Nm³/kg. coal. If there is primary and secondary air the following division has proved appropriate :—

65 per cent primary air = 5·0 Nm³/kg. coal
35 per cent secondary air = 2·6 Nm³/kg. coal.

For example, with a throughput of 80 kg. (176·36 *lbs.*) of coal per hour about 600 Nm³ per hour of air is required.

It is quite essential that the air is not drawn through by natural chimney draught as in the old-fashioned semi-gas furnaces but that it comes through by fan produced pressure under the grate whereby a static pressure of 250 to 400 mm. (9·842 *to* 15·75 *inches*) water gauge should obtain. It has been shown time and time again that combustion by simple chimney draught is altogether unfavourable. With simple plain bar grates and without pressure, up to 30 per cent or more unburnt coal is found in the ash as a result of bad air mixing.

The chimney draught should be so dimensioned, and controllable by means of a damper that the furnace neither draws air in nor blows the flames out unduly. As the quantity of burnt fuel alters and also the rate of throughput (stock) alters, the damper must be positioned accordingly. A suitable instrument for controlling correct dampering is a sensitive draught gauge in the flue between the furnace and the damper with the dial situated on the furnace " standing " (+10 to — 10 mm. water gauge). One can then prescribe a definite draught which must be maintained by appropriate dampering. As the amount of burnt fuel on the hearth alters so will the draught in the flue alter. With appropriate operation of the damper (which should best be done by means of an electric push-button operation from the furnace standing) the draught can be kept constant.

The essence of semi-gas firing consists in the use of secondary air. This is introduced through the roof over the hearth edge of the fire bridge under an angle of 35 deg. (up to 45 deg.). The exit velocity of this secondary air should be about 15 m. per sec. (49·21 *ft./sec.*) (normalized to 0 deg.) so that the volume can be calculated. It should be distributed if possible evenly over the whole width of the roof and for this reason 10 to 14 openings of an appropriate diameter are suggested. The slope of the step hearth shown is about 55 deg. It is absolutely essential that both primary and secondary air be measured. Primary air is introduced through two vertical ducts left and right underneath the grate. Into these vertical ducts measuring screens should be built. Steam or water spray should be used to cool the grate and granulate the ash. When steam is used this should be blown into the primary air several metres before the air enters. In order to get oversight of the quantity of steam introduced it is advisable to measure the steam air mixture with a simple mercury thermometer in the air-duct. Further one should also measure the air pressure under the grate with a simple U-tube so that one can watch the resistance of the fuel bed to the air-stream and be aware of clinkering in good time. The secondary air must also

be measured by means of measuring screens and simple U-tube or some other way.

The furnaceman must regulate the fuel input and firing in accordance with the quantity of primary air shown, having regard to the varying charge (throughput) conditions. The measuring of secondary air is necessary for the control of combustion and a continual oversight of the relationship of the primary to the secondary air. It has been shown to be of great advantage to arrange to have the controls for air and steam, damper, and also the flow and pressure gauges all situated directly at the firing end of the furnace standing so that the furnaceman can at all times have oversight of the instruments and be within reach of the controls.

Pre-heating of the secondary air to some 250 to 350 deg. by means of a needle or tubular recuperator increases the economy.

If all the air is blown under the grate as primary air then there is direct combustion, i.e. the fuel is not gassified on the grate but burnt. This results in the highest temperature being reached in the grate itself, which produces difficulties in ashing, etc., because the ash often becomes liquid and produces large lumps of clinker. If secondary air is available but its supply not controlled and further is supplied at a low velocity, the result is excessive smoking after firing and the production of gas from freshly supplied coal. If too much air is introduced the temperature falls and the heating efficiency of the furnace is reduced.

If one does not set out to watch the effect upon each other of fuel consumption, the supply of primary and secondary air, and dampering, then the result is uneven combustion and excessive fuel consumption. It is this fact (i.e. that the effect of these three factors upon each other is usually not controlled) which is the real reason why semi-gas furnaces are often worked with direct combustion in place of the much more economical and suitable semi-gas firing on the lines of a gas producer. Frequent and thorough rabbling is essential ; also frequent fuel supply in small quantities. Improvements with semi-gas firing are, therefore in the first place to be sought in the method of firing. One should aim at working with a high fuel bed and some 60 to 65 per cent primary air under the grate so that gas is produced, which is then burnt on the hearth with cold, or better, pre-heated air of some 35 to 40 per cent of the total air supply. At the same time there must be the right relationship between throughput and fuel supply, primary and secondary air, and dampering.

Constructional alterations on the furnace are mainly concerned with the correct introduction of secondary air ; at the same time suitable coal is essential. The height of the fuel bed as well as the proportion of primary and secondary air depend on the type of coal.

RECOMMENDATIONS

For efficient operation of semi-gas furnaces it is essential that there is a suitable furnaceman, especially with large furnaces. One must not attempt to save money by unsuitable or inadequate manning. This is shown in the following example :—

With furnace production of 10 tons per hour, coal consumption with imperfect furnace operation is about 10 per cent, i.e. one ton per hour. Even a conservative calculation of economy possible here is 25 per cent. With smaller furnaces where there is less possibility of economy, it is often practicable to have one furnaceman controlling several furnaces.

The point is to give the furnaceman clear instructions which will enable him to work the furnace efficiently. In the first place he must clearly understand the conditions of combustion and their connection with the load on the furnace.

As a result of much experience and experimental work or many furnaces of this type, the following operating instructions have been found to be a useful practical guide :—

1. FUEL.

   (a) *Quantity*. The maximum permissible fuel consumption for any given furnace throughput of the various types of steel must be clearly shown on a diagram at the furnace.

   (b) *Firing*. When the coal cannot be supplied by means of an independent system such as belt or bucket conveyers, but is supplied by hand shovelling, then firing must at all times take place in small quantities, e.g. two shovelsful every three minutes instead of two large feedings per hour. With large shovellings the quantity of gas pro-

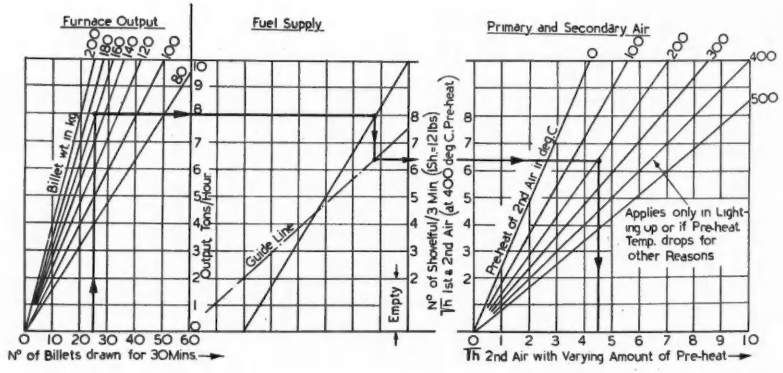

Furnace Output
Fuel Supply
Primary and Secondary Air
Billet wt. in kg
Output Tons/Hour
Guide Line
Empty
N° of Shovelful/3 Min. (Sh.=12 lbs)
Th 1st & 2nd Air (at 400 deg C. Pre-heat)
Pre-heat of 2nd Air in deg C.
Applies only in Lighting up or if Pre-heat Temp. drops for other Reasons
N° of Billets drawn for 30Mins.
Th 2nd Air with Varying Amount of Pre-heat

Fig. 2. Regulation of air supply in relation to throughput on semi-gas furnace.

duced rises very considerably after the fuel has been introduced so that in order to burn this quantity of gas economically, both the air supply and the chimney damper must be altered. Thereupon, as the quantity of gas slowly diminishes during the course of gasification, the quantity of secondary air and the dampering must be slowly altered accordingly, otherwise there will be an excess of air, resulting in excessive scaling of the stock. If, however, the fuel supply is so regulated as to be done in small quantities at frequent intervals, one approaches the conditions of continuous fuel supply, and the controls need not be continually altered. This method of operation is, therefore, preferable in most cases.

2. AIR SUPPLY.

The quantity of air, separated into primary and secondary air, must be clearly stipulated for the type of fuel and the desired furnace performance. As any given furnace performance requires the appropriate quantities of fuel, primary air and secondary air, it is most useful to develop a diagram for this purpose from which the furnaceman can readily read off these quantities. (See Fig. 2.)

3. TEMPERATURE.

The working temperatures separated into furnace temperature and drawing temperature of the stock are to be stipulated. The temperature of the exit gases must not exceed 800 deg. or at the most 900 deg. C. with the usual type of needle recuperator. It is useful to have a warning signal but at the least a pyrometer for the exit gases is necessary.

4. DAMPERING.

The dampering is to be so controlled that the furnace neither blows the flames out nor draws cold air in. During any stoppages the damper must be completely closed.

5. STOPPAGES.

The furnaceman must always see to it that by careful feeding in small quantities the furnace is filled with a slow circulating flame during any stoppages to prevent burning of the stock. Only shortly before the end of the stoppage must primary and secondary air and damper as well as coal supply be brought back to normal.

6. CLEANING OF THE GRATE.

A further primary condition for the correct working of the semi-gas furnace is thoroughly cleaning the grate. The de-clinkering or ashing must be carried out from the grate very carefully and this must be thoroughly inculcated upon the furnaceman. For cleaning the step grate, a dummy grate is frequently used, made (say) of scrap bars. When the ash door has been opened the dummy grate is inserted on the third bottom step of the grate and pushed through to the far wall of the fire. Thereupon the space between the dummy grate and the bottom grate is emptied. Then the bottom grate is re-inserted and the dummy grate is pulled out. The upper bed is then rabbled and the ash therein falls through and is cleared out at the next cleaning.

7. STEAM SUPPLY.

This is dependent on the ash zone and can be up to 50 per cent of the amount of coal. The furnaceman must be instructed as to what steam/air temperature to keep to.

8. CONTROL FIGURES.

In order to control the fuel consumption the following figures should be kept :—

Furnace Output	: Tons per hour.
Hearth Performance	: Tons per sq. ft. hearth area per hour.
Coal Consumption	: Tons per working hour. Tons per 24 hours or per shift (including stoppages, etc). Tons per ton input.
Heat Supply	: B.T.U's per ton input.

It has been found most beneficial to give the furnacemen a short set of instructions covering their duties and responsibilities. (In order that these can really be carried out it is, of course, absolutely necessary that the furnacemen have sufficient instruction in the first place). An example of such operating instructions is given below:

" The forehand furnaceman is responsible for the efficient working of the furnace, the correct control of primary and secondary air, and the appropriate rate of firing, having regard to the furnace loading and working at any given time. The correct quantity of primary and secondary air as well as the number of shovelsful of coal are to be read off the diagram every half hour and compared with the furnace-load. The forehand must tell the under-hand the number of shovelsful per three minutes and the under-hand must work to this. During stoppages air and coal supply are to be shown against the " empty " line on the diagram.

Further, the forehand must watch :—

(i) BILLET TEMPERATURE. If the billets are too cold he must give instructions for extra firing and for appropriately greater air supply. If the billet temperature is too high the reverse must be carried out.

(ii) FURNACE FLAME. The flame must be yellowish and reach right to the flue end. If the flame is short and transparent, extra firing is required, and if a thick smoky flame, extra secondary air is required.

(iii) PRESSURE UNDER FURNACE. This is to be controlled so that there is slight flaming at the doors.

(iv) EXIT GAS TEMPERATURE. If the temperature exceeds 850 deg. C. the entry damper of the recuperator must be open and the hot gas damper to the recuperator closed accordingly.

(v) Steam air temperature is to be kept at. . . . deg C.

(vi) Half an hour before the end of the shift no more coal must be supplied so that the coal burns out. The primary air supply is not to be altered during this time. Secondary air and flue draught must be reduced appropriately as the gas supply is reduced.

(vii) CLEANING OF GRATE. When the ash doors have been opened the under-hand must satisfy himself that the steam supply (or water spray) is functioning correctly. When the grate has been cleaned it must be so worked that in about half an hour the normal height of the fuel bed has again been reached. When this has been obtained the forehand must give the under-hand the appropriate fuel supply figure (number of shovels per 3 minutes as per diagram).

SWITZERLAND

THEORETICAL CONSIDERATIONS OF GAS TURBINE AND JET PROPULSION FOR AIRCRAFT.

By K. R. SCHEUTER. (From *Flugwehr und-Technik*, Vol. 7, No. 8, August, 1945, pp. 209-214, 11 illustrations).

THE purpose of this article is to investigate the efficiency of the simplest thermodynamic cycle suitable for Aircraft power plants, and, in particular, to examine the question of what percentage of the available heat drop to utilize in the jet and propeller respectively. In the following " power " stands for " thrust power," i.e. the product of thrust and speed.

Fig. 1. shows the plant and Fig. 2. the corresponding entropy chart. The arrangement of the plant differs from the pure jet propulsion unit by the addition of a turbine-driven propeller. Thus, the turbine supplies power to drive both propeller and compressor.

For reasons of simplicity, the specific heat is taken to be constant ($c_p = 0{\cdot}24$ Kcal / Kg deg. C). This assumption is permissible when dealing with air.

The temperature drop $T_3—T_7 = T_4—T_2$ is consumed by the compressor ; the drop $T_7—T_9$, also utilized in the turbine, is consumed by the propeller ; the remainder of the drop $T_9—T_{11}$ goes into the jet.

The following are the fundamental equations : (see : Remarks on Jet-Propulsion of Aircraft, by J. Ackeret, THE ENGINEERS' DIGEST, Vol. 5, p. 221).

At the duct inlet the air speed w_1 is decelerated to $w_2 = \alpha\, w_1$. The temperature increase due to ram effect is

$$T_2—T_1 = \frac{A}{2g\, c_p}(w_1^2—w_2^2) = \frac{A}{2g\, c_p}(1—\alpha^2)\, w_1^2$$

Substituting for the speed of sound $a_1 = \sqrt{gk\, RT_1}$ and the Mach number $M_1 = \frac{w_1}{a_1}$, we obtain

$$\frac{T_2}{T_1} = 1 + \frac{k-1}{2}(1-\alpha^2)\, M_1^2 = \mathfrak{M} \quad .. \quad (1)$$

Thus, for the air speed

$$w_1 = \sqrt{\frac{2g\, c_p}{A}\,\frac{\mathfrak{M}-1}{1-\alpha^2}}\, T_1 \quad .. \quad .. \quad (1a)$$

The adiabatic temperature ratio due to compression is

$$\frac{T_3}{T_2} = \delta = \left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k}}, \text{ and } \frac{T_3}{T_1} = \delta\,\mathfrak{M} \quad .. \quad (2)$$

Let the adiabatic efficiency of the compressor be η_K ; then

$$T_4 - T_2 = \frac{T_3 - T_2}{\eta_K}; \quad \frac{T_4}{T_1} = \mathfrak{M}\left(1 + \frac{\delta - 1}{\eta_K}\right) \quad (3)$$

In the combustion chamber the air is heated from T_4 to T_5. We denote

$$T_5 = KT_1 \quad .. \quad .. \quad .. \quad (4)$$

If adiabatic expansion would take place down to atmospheric pressure, the temperature T_{10} would be reached. Because of expansion between equal pressures, we have

$$\frac{T_{10}}{T_1} = \frac{T_5}{T_3} = \frac{K}{\delta\,\mathfrak{M}} \quad .. \quad .. \quad (5)$$

The part of the turbine output used for driving the compressor is expressed by

$$T_4 - T_2 = T_5 - T_7 \text{ or } \frac{T_3 - T_2}{\eta_K} = \eta_T\,(T_5 - T_6)$$

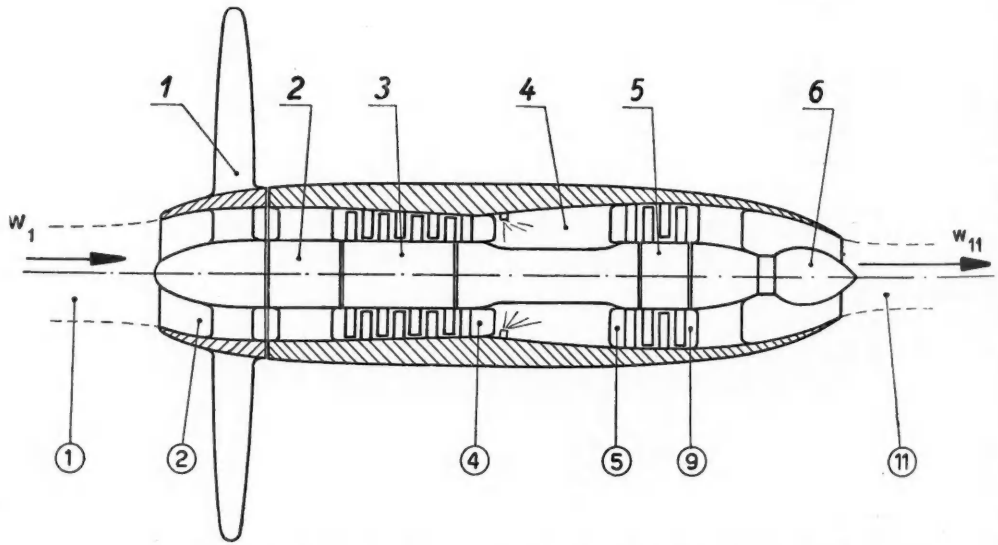


Fig. 1. Propeller-jet propulsion. Numerals in circles refer to state in the entropy chart (Fig. 2).

(1) propeller, the hub of which forms a ducted fan.
(2) reduction gear
(3) axial-flow compressor
(4) combustion chamber
(5) gas turbine
(6) movable bullet for control of discharge area.

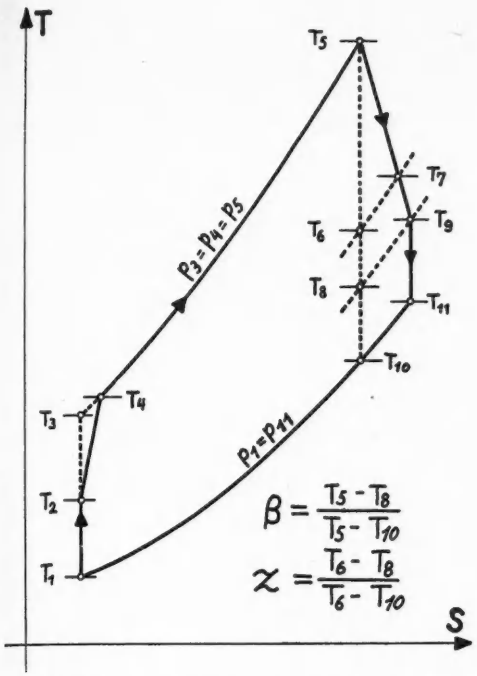


Fig. 2. **Entropy chart of process according to Fig. 1.**

Herefrom,

$$\frac{T_8}{T_1} = K - \mathfrak{M}\,\frac{\delta - 1}{\eta_K\,\eta_T} \quad .. \quad .. \quad (6)$$

$$\frac{T_7}{T_1} = K - \mathfrak{M}\,\frac{\delta - 1}{\eta_K} \quad .. \quad .. \quad (7)$$

Further expansion in the turbine to T_9, or to T_8 adiabatically, produces the power absorbed by the propeller. Let the temperature T_8 be defined by the equation

$$T_5 - T_8 = \beta\,(T_5 - T_{10}) \quad .. \quad .. \quad (8)$$

Therefore, from equations (4), (5) and (8)

$$T_5 - T_9 = \eta_T\,(T_5 - T_8) = \eta_T\,\beta\,(T_5 - T_{10}) \quad .. \quad (9)$$

$$= \eta_T\,\beta\left(K - \frac{K}{\delta\,\mathfrak{M}}\right) T_1 = \eta_T\,\beta\,K\phi\,T_1$$

where $\phi = \dfrac{\delta\mathfrak{M} - 1}{\delta\,\mathfrak{M}}$.

With $T_5 - T_7 = \mathfrak{M}\,\dfrac{\delta - 1}{\eta_K}\,T_1$ and equations (4) and (7) we obtain the drop absorbed at the propeller shaft using also eq. (9)

$$T_7 - T_9 = \left(\eta_T\,\beta\,K\phi - \mathfrak{M}\,\frac{\delta - 1}{\eta_K}\right) T_1 \quad .. \quad (10)$$

The jet energy is due to the drop $T_9 - T_{11}$. Because of expansion between equal pressures, the following relation holds good.

$$\frac{T_9 - T_{11}}{T_8 - T_{10}} = \frac{T_9}{T_8} \quad .. \quad .. \quad (11)$$

From equations (4), (5) and (8) it follows :

$$T_8 - T_{10} = (1-\beta)\,(T_5 - T_{10}) = (1-\beta)\,K\phi\,T_1 \quad (12)$$

$$T_8 = T_5 - \beta\,(T_5 - T_{10}) = K\,(1 - \beta\phi)\,T_1 \quad .. \quad (13)$$

From equations (4), (5) and (9)

$$T_9 = T_5 - \eta_T\,\beta\,(T_5 - T_{10}) = K\,(1 - \eta_T\,\beta\,\phi)\,T_1 \quad (14)$$

Thus, the temperature drop in the jet is

$$T_9 - T_{11} = \frac{1 - \eta_T\,\beta\phi}{1 - \beta\phi}\,(1 - \beta)\,K\phi\,T_1 \quad .. \quad (15)$$

In addition to this drop, there is a drop ΔT due to the kinetic energy of speed w_2 which, for reasons of simplicity, is taken to be constant up to turbine discharge. Using eq. (1a)

$$\Delta T = \frac{A w_2^2}{2g\,c_p} = \frac{A\,\alpha^2\,w_1^2}{2g\,c_p} = \frac{\alpha^2}{1-\alpha^2}\,(\mathfrak{M} - 1)\,T_1 \quad (16)$$

The discharge velocity of the jet, therefore is

$$w_{11} = \sqrt{\frac{2g\,c_p}{A}\left[(T_9 - T_{11}) + \Delta T\right]} =$$

$$= \sqrt{\frac{2gc_p}{A}\left[K\phi\,\frac{1-\eta_T\beta\phi}{1-\beta\phi}\,(1-\beta) + \frac{\alpha^2}{1-\alpha^2}\,(\mathfrak{M}-1)\right] T_1} \quad (17)$$

According to the impulse theorem the thrust for each kg/sec. of air is

$$s = \frac{1}{g}\,(w_{11} - w_1),$$

and the power in heat units

$$N_s = A.s.w_1 = \frac{A}{g}\,w_1^2\left(\frac{w_{11}}{w_1} - 1\right).$$

The propeller thrust power is given by

$$N_p = \eta_p\,c_p\,(T_7 - T_9)$$

where η_p is the propeller efficiency.

The heat supplied is

$$Q = c_p\,(T_5 - T_4)$$

The efficiency of the cycle when related to thrust power is

$$\eta = \frac{N_p + N_s}{Q}$$

Substituting the known values, we find by means of equations (1a), (3), (4), (10) and (17)

$$\eta = \frac{\eta_p\left(\eta_T\,\beta\,K\phi - \mathfrak{M}\,\dfrac{\delta - 1}{\eta_K}\right)}{K - \mathfrak{M}\left(1 + \dfrac{\delta - 1}{\eta_K}\right)} +$$

$$+ \frac{2\,\dfrac{\mathfrak{M}-1}{1-\alpha^2}\left\{\sqrt{\dfrac{K\phi\,\dfrac{1-\eta_T\beta\phi}{1-\beta\phi}\,(1-\beta)}{\dfrac{\mathfrak{M}-1}{1-\alpha^2}} + \alpha^2 - 1}\right\}}{K - \mathfrak{M}\left(1 + \dfrac{\delta - 1}{\eta_K}\right)} \quad (18)$$

The parameter β giving the ratio of the drop utilized in the turbine to the total drop available, though convenient for calculation, does not indicate directly the ratio of

propeller power to jet power. The ratio

$$z = \frac{T_6 - T_8}{T_6 - T_{10}}$$ will, therefore, be employed.

By substituting, eqs. (5), (6) and (8) yield ;

$$z = \frac{\beta K\phi - \mathfrak{M}\dfrac{\delta-1}{\eta_K\,\eta_T}}{K\phi - \mathfrak{M}\dfrac{\delta-1}{\eta_K\,\eta_T}} \quad .. \quad .. \quad (19)$$

$$\text{or} \quad \beta = \frac{z\left(K\phi - \mathfrak{M}\dfrac{\delta-1}{\eta_K\,\eta_T}\right) + \mathfrak{M}\dfrac{\delta-1}{\eta_K\,\eta_T}}{K\phi} \quad (19a)$$

The limiting values are :

$z = 0$ pure jet propulsion

$z = 1$ pure propeller propulsion.

Let us first examine the efficiency in relation to z.

Fig. 3 shows a particular example from which it be-

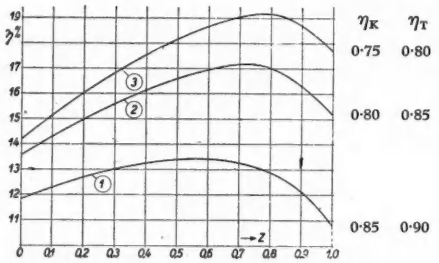


Fig. 3. **Effect of part efficiencies on total efficiency.**
$w_1 = 250$ m/sec. ; $H = 4000$ m ; $K = 4$; $\delta = 1{\cdot}3$; $\alpha = 0{\cdot}7$; $\eta_p = 0{\cdot}85$.

comes clear that

1. the propeller increases the efficiency considerably
2. the gain is the greater the better the part efficiencies are
3. with increasing part efficiencies the optimum value shifts towards increasing z.

The influence of speed on the efficiency is shown in Fig. 4. Obviously, with increasing speed the efficiency increases. This is due primarily to the increasing jet efficiency with decreasing difference between jet discharge velocity and flight speed. Also the ram effect increases with speed. The maximum temperature of the cycle is limited by the properties of heat resisting steels. The effect of temperature on efficiency is shown in Fig. 5. The present maximum value for turbine inlet temperature T_5 is given by a value $K = 3{\cdot}6$ to 4. Therefore, for all further calculations, $K = 4$ is assumed ($T_5 =$ 1048 deg. C abs., i.e. 775 deg. C). For this case the influence of δ has also been investigated (Fig. 6). As can be seen, the efficiency increases with increasing compression ratio. An increase in compression ratio necessitates an increase in the number of stages, which increases the weight of the compressor. The curves of Fig. 6 are based on a compressor efficiency $\eta_K = 0{\cdot}85$. It should be borne in mind that for each of the six curves shown this means a different stage efficiency η_{ks} assuming that for one compressor the stage efficiency is constant. For curve 1 in Fig. 6, $\eta_{ks} = 0{\cdot}866$; and Fig. 7 gives an example of how the compressor efficiency changes

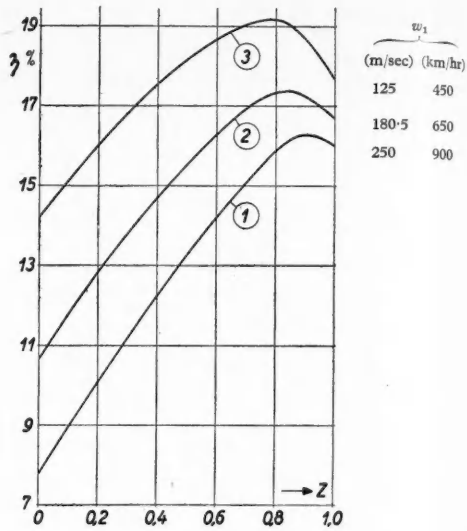


Fig. 4. **Effect of Speed on efficiency.**
$H = 4000$ m ; $\eta_p = \eta_k = 0{\cdot}85$; $\eta_T = 0{\cdot}9$; $K = 4$; $\delta = 1{\cdot}3$ $\alpha = 0{\cdot}7$.

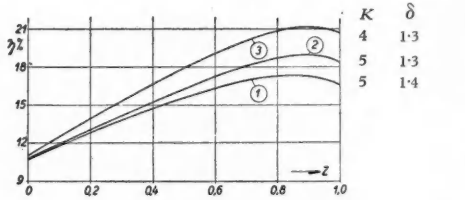


Fig. 5. **Effect of maximum temperature** ($T_5 = KT_1$) **and compression ratio.**
$w_1 = 180{\cdot}5$ m/sec ; $H = 4000$ m ; $T_1 = 262$ deg. C abs ; $\alpha = 0{\cdot}7$; $\eta_p = \eta_k = 0{\cdot}85$; $\eta_T = 0{\cdot}9$

with compression ratio at constant stage efficiency ($\eta_{ks} =$ 0·866). On this curve $\eta_{k_1} = 0{\cdot}85$ represents the compressor efficiency of curve 1, Fig. 6. If we add further stages to this compressor till $p_3/p_2 = 6{\cdot}407$ is reached (accordingly $\delta = 1{\cdot}7$) the compressor efficiency obtained would be $\eta_{k_2} = 0{\cdot}828$. To get, with this increased number of stages, an efficiency $\eta_k = 0{\cdot}85$, the stage efficiency would have to be increased to 89 per cent., a value which can, at present, not be realized.

To assess the total efficiency of the cycle for $\delta = 1{\cdot}7$, $K = 4$, the calculation has been repeated with reduced compressor efficiency (Fig. 8). The reduction of total efficiency at $\delta = 1{\cdot}7$ is smaller than 1 per cent. when compared with the curve based on 85 per cent. compressor efficiency.

Since $\eta = \dfrac{N_p + N_s}{Q}$, the thrust for each kg/sec. of air is $s = \dfrac{N_p + N_s}{A\,w_1} = \dfrac{\eta\,Q}{A\,w_1}$

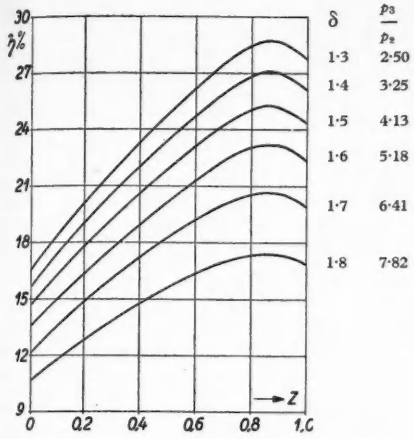


Fig. 6. **Effect of pressure ratio on efficiency.**
$w_1 = 180{\cdot}5$ m/sec ; $H = 4000$ m ; $K = 4$; $\alpha = 0{\cdot}7$;
$\eta_p = \eta_k = 0{\cdot}85$; $\eta_T = 0{\cdot}90$

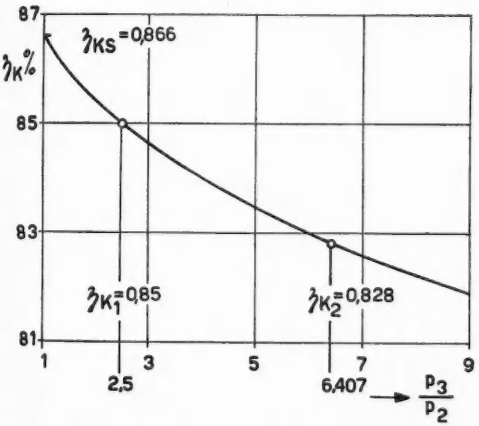


Fig. 7. **Theoretical curve for compressor efficiency as a function of pressure ratio.**

$$\eta_k = \frac{\left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k}} - 1}{\left(\frac{p_3}{p_2}\right)^{\frac{k-1}{k \cdot \eta_{ks}}} - 1}$$

Substituting the known values, and for the speed of sound $a_1 = \sqrt{gk\ RT_1}$, we get

$$s = \frac{a_1}{2g}\sqrt{\frac{2}{k-1}}\ \frac{\eta\left[K - \mathfrak{M}\left(1 + \frac{\delta-1}{\eta_k}\right)\right]}{\sqrt{\frac{\mathfrak{M}-1}{1-\alpha^2}}} \qquad (20)$$

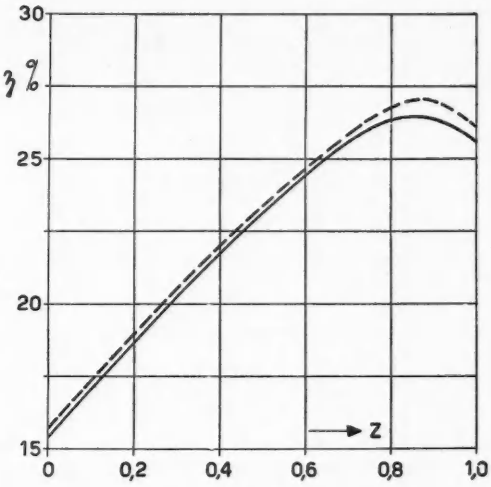


Fig. 8. **Efficiency of cycle with $\delta = 1{\cdot}7$, K = 4 when taking into account the decrease of compressor efficiency. The dotted curve corresponds to curve 5 in Fig. 6.**

and after substituting for η :

$$s = \frac{a_1}{2g}\sqrt{\frac{2}{k-1}}\left\{\frac{\eta_p\left(\eta_T \beta K\phi - \mathfrak{M}\frac{\delta-1}{\eta_k}\right)}{\sqrt{\frac{\mathfrak{M}-1}{1-\alpha^2}}} + \right.$$

$$\left. +2\left[\sqrt{\frac{K\phi(1-\eta_T\beta\phi)\ (1-\beta)}{1-\beta\phi}} + \left(\frac{\mathfrak{M}-1}{1-\alpha^2}\right)\alpha^2 - \sqrt{\frac{\mathfrak{M}-1}{1-\alpha^2}}\right]\right\} \qquad (20a)$$

The total thrust S is proportional to the mass flow of air G,

$$S = G \,.\, s \text{ (kg)} \quad .. \quad .. \quad (21)$$

As equation (20) shows,

$$s\,(\beta) = \eta\,(\beta) \text{ constant}$$

The tendency of the thrust and efficiency curves is the same (Fig. 9). The same applies to the thrust power curve. In the theoretical derivations the losses in compressor and turbine have been considered only. The losses at the duct inlet, the combustion chamber and at jet discharge remained unaccounted for. To take these into account, the entropy chart has been redrawn accordingly (Fig. 10). The ram at the duct inlet raises the temperature to $T_2 = \mathfrak{M}T_1$, but the corresponding adiabatic pressure p_2 is not obtained because of the pressure loss Δp_2. To the smaller pressure p_2' corresponds a temperature T_2' which can be expressed as

$$\frac{T_2}{T_2'} = \left(\frac{p_2}{p_2'}\right)^{\frac{k-1}{k}} = \left[\frac{1}{1 - \frac{\Delta p_2}{p_2}}\right]^{\frac{k-1}{k}} = \left(\frac{1}{1-\epsilon_1}\right)^{\frac{k-1}{k}} = \xi_1$$

and herefrom $\qquad \frac{T_2'}{T_1} = \frac{\mathfrak{M}}{\xi_1}$

The efficiency of compression due to ram effect can be expressed as

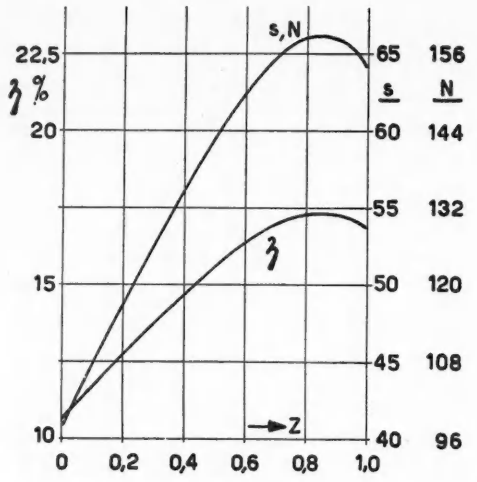


Fig. 9. **Efficiency, thrust and thrust power for mass flow per second.**

$$s = \left[\frac{\text{kg}}{\text{kg/sec}}\right]; \quad N = \frac{s w_1}{75} \left[\frac{\text{HP}}{\text{kg/sec}}\right]$$

$w_1 = 180{\cdot}5$ m/sec ; $H = 4000$ m ; $K = 4$; $\delta = 1{\cdot}3$; $\alpha = 0{\cdot}7$
$\eta_p = \eta_k = 0{\cdot}85$; $\eta_T = 0{\cdot}90$.

$$\eta_e = \frac{\Delta T_{ad}}{\Delta T_{actual}} = \frac{T_2' - T_1}{T_2 - T_1}$$

Therefore
$$\eta_e = \frac{\frac{\mathfrak{M}}{\xi_1} - 1}{\mathfrak{M} - 1}$$

or
$$\xi_1 = \frac{\mathfrak{M}}{1 + \eta_e (\mathfrak{M} - 1)}$$

If no further losses occur, the temperature T_{10}' at atmospheric pressure would be reached by adiabatic expansion from T_5.

Because of expansion between equal pressures we have

$$\frac{T_2}{T_2'} = \frac{T_{12}}{T_{12}'} = \xi_1$$

or
$$T_{12}' = \frac{T_2}{\xi_1}$$

For the same reason

$$\frac{T_{10}'}{T_{12}} = \frac{T_{10}}{T_{12}'} = \xi_1 \cdot \frac{T_{10}}{T_{12}}$$

or
$$T_{10}' = \xi_1 T_{10} \quad .. \quad .. \quad (22)$$

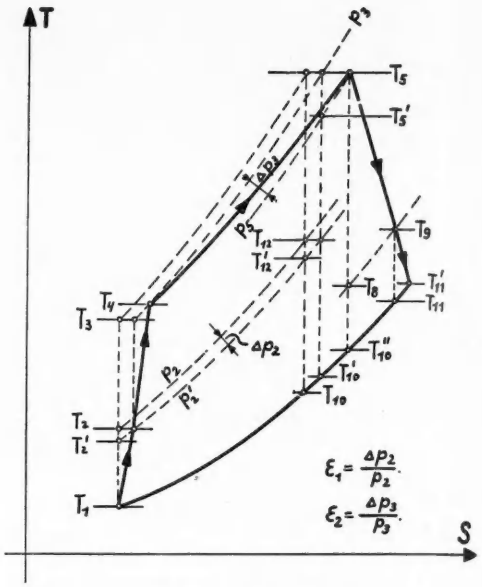


Fig. 10. **Entropy chart of the process taking account of losses.**

In the combustion chamber the pressure decreases by Δp_3 from p_3 to p_5. Then

$$\epsilon_2 = \frac{\Delta p_3}{p_3} = 1 - \frac{p_5}{p_3}$$

Again

$$\frac{T_5}{T_5'} = \left(\frac{p_3}{p_5}\right)^{\frac{k-1}{k}} = \left(\frac{1}{1 - \epsilon_2}\right)^{\frac{k-1}{k}} = \xi_2$$

$$\frac{T_{10}''}{T_5} = \frac{T_{10}'}{T_5'} = \xi_2 \cdot \frac{T_{10}'}{T_5}$$

$$T_{10}'' = \xi_2 \cdot T_{10}'$$

or
$$T_{10}'' = \xi_1 \cdot \xi_2 \cdot T_{10} \quad .. \quad .. \quad .. \quad (23)$$

Thus, the adiabatic drop in the jet is reduced to

$$T_9 - T_{11} = (T_8 - \xi_1 \cdot \xi_2 \cdot T_{10}) \frac{T_9}{T_8} \quad .. \quad .. \quad (15a)$$

With the efficiency of expansion at jet discharge

$$\eta_a = \frac{T_9 - T_{11}'}{T_9 - T_{11}}$$

the actual drop in the jet is

$$T_9 - T_{11}' = \eta_a (T_8 - \xi_1 \cdot \xi_2 \cdot T_{10}) \frac{T_9}{T_8} \quad .. \quad (15b)$$

where T_8, T_9 and T_{10} are the temperatures of the ideal cycle without losses.

Since all temperatures are given, the equations for thrust and efficiency can be written down.

CLOSING TIME OF JOHNSON-TYPE NEEDLE VALVES AT SILLRE HYDROELECTRIC POWER STATION.

By R. KARLSON. (From *Teknisk Tidskrift*, Vol. 75, No. 47, November 24, 1945, pp. 1283-1288, 9 illustrations.)

THE Sillre hydroelectric plant is a high head plant of unusual design insofar as each of the two water wheels is directly coupled with a two-stage centrifugal pump (Fig. 1). The water wheels are designed for a capacity of 8,580 KW each at 600 r.p.m. With a design head of 190 metres, the throughput per turbine is 5·4 cu. m. per second. Each pump is designed for a delivery of 2·4 cu. m. per second against a head of 200 metres, the power consumption being 5,700 KW per pump. When pumping, the generators are operated as motors, and the water-wheels are idling. Vice versa, the pumps are idling when the sets are generating.

The Johnson type needle valves (Fig. 2) controlling the pump delivery are so designed that they can also serve as automatic non-return valves. The closing time of these valves is of importance insofar as they must go into action in the event of a sudden cessation of pump delivery. The closing time of the valve must be neither too short nor too long. If it is too short, a vacuum will form under the valve, and, as the valve closes, water hammer will result, hereby incurring the risk of destruction of the valve or of the penstock. On the other hand, an unduly long closing time will result in a reversal of flow which will cause the pump to operate as water wheel. Operation of the Johnson valve is carried out by means of pressure-water taken from the penstock of the turbines and controlled by a number of small valves. The latter can be electrically operated from a control board at will, but they will also go into action automatically in the event of a plant breakdown.

A previous method of computing closing times was based upon the use of difference equations, but this method is most tedious, as relatively small intervals must be considered. By means of the method described below, the water velocity in the penstock is expressed as a function of time, and the actual computation can be carried out easily.

In the case of a cessation of pumping, the retarding force in the pipe line is

$$[H+H_f - H_m]\gamma . F$$

where H is the static head in m. water column, H_m is the manometric pressure head of the pump in m. water column, H_f are the losses in m. water column, γ is the specific gravity of the water, equal to 1 gram per cu. cm.

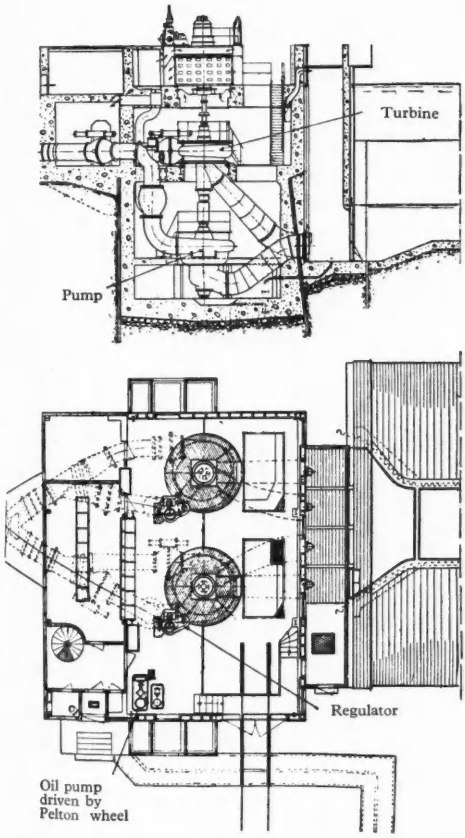


Fig. 1. **Section through Sillre hydroelectric power station.**

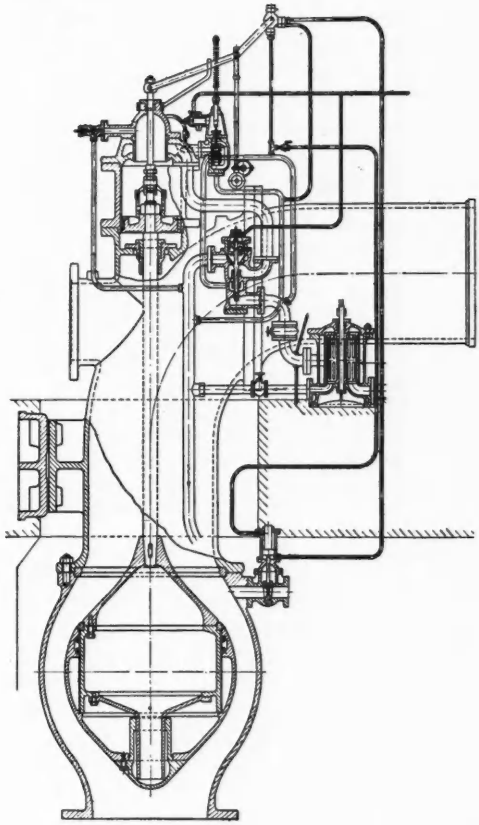


Fig. 2. **Section through Johnson Valve.**

The decelerating mass of the water in the pipe is

$$\frac{\gamma.L.F}{g}$$

where L is the length of the pipe, and g is the gravitational constant. The deceleration is given by dc/dt, where c is the water velocity and t is the time in seconds. As it is

force = mass × acceleration, it will be

$$[H+H_f—H_m]\gamma.F = \frac{\gamma.L.F}{g}\left(-\frac{dc}{dt}\right)$$

$$\text{so that } dc/dt = \frac{g}{L}[H_m—H—H_f] \qquad (1)$$

This differential equation cannot be integrated in its present form, as H_m and H_f are functions of the time. This functionality can however be found from the interrelationship existing between the delivery rate Q, the power absorption N, and the speed n of the pump. It is $Q/Q_o=n/n_o$, where Q_o and n_o represent the delivery rate and the pump speed respectively under normal operating conditions. As it is also : $Q=c\,.\,F$, it will be

$$c\,.\,F/Q_o=n/n_o$$

By introducing the angular speed ω, it can be written

$$c=c_o\,\frac{\omega}{\omega_o} \qquad .. \quad .. \qquad (2)$$

The pressure at the delivery side of the pump varies according to the relationship

$$\frac{H_m}{H_{mo}} = \left(\frac{n}{n_o}\right)^2 = \left(\frac{\omega}{\omega_o}\right)^2$$

$$\text{or} \qquad H_m = H_{mo}\left(\frac{\omega}{\omega_o}\right)^2 \qquad .. \quad .. \quad .. \qquad (3)$$

where H_{mo} is the manometric pressure at the pump outlet in metres under normal operating conditions. For the revolving mass of the pump it is

$$M = -\mathcal{J}\frac{d\omega}{dt} \qquad .. \quad .. \quad .. \qquad (4)$$

where M is the retarding moment and $\mathcal{J}$ is the moment of inertia in m.kg.sec.^{2} The power absorption of the pump varies according to the relationship

$$\frac{N}{N_o} = \left(\frac{n}{n_o}\right)^3 = \left(\frac{\omega}{\omega_o}\right)^3, \text{ or } \frac{M}{M_o} = \left(\frac{\omega}{\omega_o}\right)^2 \qquad (5)$$

By introducing this expression for M into formula (4), it follows :

$$M_o\left(\frac{\omega}{\omega_o}\right)^2 = -\mathcal{J}\frac{d\omega}{dt}$$

$$\text{or} \qquad \frac{M_o}{\mathcal{J}}\,.\,\frac{1}{\omega_o{}^2}\,dt = -\frac{d\omega}{\omega^2}$$

By integration it becomes :

$$\frac{M_o}{\mathcal{J}}\,.\,\frac{1}{\omega_o{}^2}\,.\,t = \frac{1}{\omega} + K$$

where K is a constant. As for $t=0$ it is $\omega=\omega_o$, it will be $0=1/\omega_o+K$; $K=-1/\omega_o$. By introducing this value for K into the aforementioned equation, it becomes

$$\frac{M_o}{\mathcal{J}}\,\frac{1}{\omega_o{}^2}\,.\,t = \frac{1}{\omega} - \frac{1}{\omega_o}$$

and by solving for ω it is found

$$\frac{1}{\omega} = \frac{M_o t+\mathcal{J}\omega_o}{\mathcal{J}.\omega^2{}_o}$$

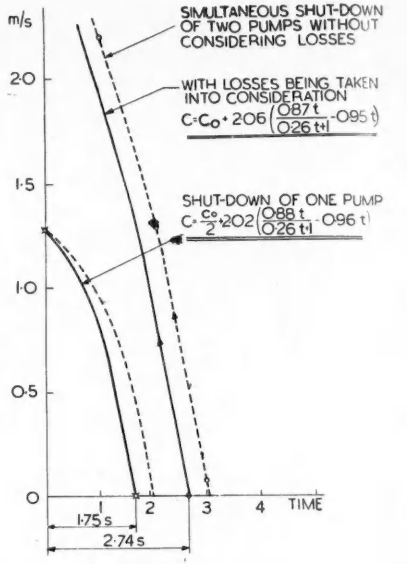


Fig. 3. Water velocity in penstock at time t after shutting down one or two pumps.

and

$$\omega^2 = \frac{\mathcal{J}^2.\omega^4{}_o}{(M_o t+\mathcal{J}\omega_o)^2} \qquad .. \quad .. \quad .. \quad (6)$$

From the equations (1), (3), and (6), the following differential equation can be established, to wit :—

$$\frac{dc}{dt} = \frac{g}{L}\left[\frac{H_{mo}}{\omega_o{}^2}\,\frac{\mathcal{J}^2.\omega_o{}^4}{(M_o.t+\mathcal{J}\omega_o)^2} - H - H_f\right]$$

This can also be written

$$\frac{dc}{dt} = \frac{gH_{mo}}{L}\left[\frac{\mathcal{J}^2\omega_o{}^2}{(M_o.t+\mathcal{J}\omega_o)^2} - \frac{H}{H_{mo}} - \frac{H_f}{H_{mo}}\right] \qquad (7)$$

This differential equation can be solved by neglecting the losses H_f, so that

$$dc = \frac{g.H_{mo}}{L}\left[\frac{\mathcal{J}^2\omega_o{}^2}{(M_o.t+\mathcal{J}\omega_o)^2} - \frac{H}{H_{mo}}\right]dt$$

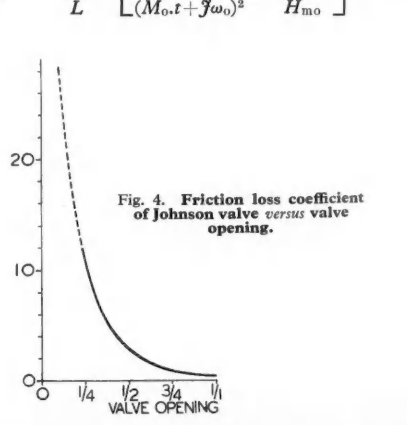


Fig. 4. Friction loss coefficient of Johnson valve *versus* valve opening.

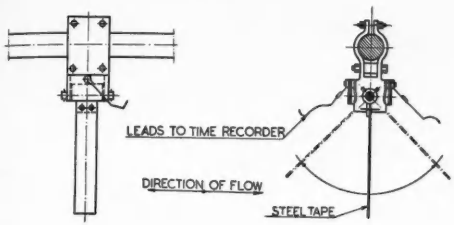


Fig. 5. Flow direction indicator.

and by integration :

$$\int_{c=c_0}^{c} dc = \frac{g.H_{mo}}{L} \int_{t=0}^{t} \left[\frac{\mathcal{J}^2\omega^2{}_o}{(M_o.t+\mathcal{J}\omega_o)^2} - \frac{H}{H_{mo}} \right] dt$$

This yields the expression

$$c = c_o + \frac{g.H_{mo}}{L} \left[\frac{t}{\frac{M_o \,.\, t}{\mathcal{J} \,.\, \omega_o} + 1} - \frac{H \,.\, t}{H_{mo}} \right] \qquad (8)$$

This equation gives the water velocity c which prevails after t seconds. By applying this method of analysis to the conditions prevailing at the Sillre station, the following result is obtained.

With one of the two pumps being in operation and delivering a maximum water delivery of $Q = 2{,}720$ litres per second, it is $\Sigma l.c = 1200$ m² per second. The total length of the penstock is approx. 950 metres so that

$c_o = \dfrac{\Sigma l.c}{L} = \dfrac{1{,}200}{950} = 1{\cdot}27$ m. per sec. With two pumps in operation, it is $c_o = 2{\cdot}54$ m. per sec. The moment of inertia $\mathcal{J}$ of the complete set is $\mathcal{J} = \dfrac{G{\cdot}D^2}{4g}$, where the flywheel effect $G{\cdot}D^2 = 25{,}000$ kg.m². It is therefore $\mathcal{J} = \dfrac{25{,}000}{4g} \approx 630$ m.kg.sec². The following values were obtained by test (with one pump being in operation).

$H = 195{\cdot}6$ m. water column, $N_o = 8{,}100$ hp.
$H_{mo} = 188{\cdot}5$ m. „ „ $n_o = 600$ r.p.m.

If two pumps are in operation, the manometric pressure H_{mo} is increased by approx. 3·5 m. It is further :

$$M_o = 716{\cdot}2\frac{N_o}{n_o} = 716{\cdot}2\frac{8{,}100}{600} = 9{,}650 \text{ kg.m.}$$

and $\omega_o = \dfrac{\pi\, n_o}{30} = \dfrac{\pi\, 600}{30} = 62{\cdot}8$ radians per sec.

By introducing these values into equation (8), the following correlated values for c and t are found to obtain :

	t =	0	1	2	3	
One pump :	c_1 =	1·27	0·93	0·06	—	m. per sec.
Two pumps :	c_2 =	2·54	2·15	1·34	0·11	m. per sec.

This relationship is charted in Fig. 3. The friction losses incurred in the penstock can be found from the expression $h_f = \dfrac{c^2 L}{k^2 R}$, where L is the pipe length, R is the hydraulic radius $= D/4$ m., k is given by $k = K\,(1+0{\cdot}5\sqrt{R})$, and c is the velocity of flow. For $L = 950$ m., $R = 1{\cdot}85/4$, and $K = 60$, it is

$$h_{f1} = 0{\cdot}32\, c^2{}_{01} \left(\frac{\omega}{\omega_0}\right)^2$$

The loss in head incurred in the needle valves is

$$h_{f2} = \zeta_2 \frac{{c_0}^2{}_2}{2\,.\,g} \left(\frac{\omega}{\omega_0}\right)^2$$

and that incurred in the 90 deg. bends is

$$h_{f3} = \zeta_3 \frac{{c_0}^2{}_3}{2\,.\,g} \left(\frac{\omega}{\omega_0}\right)^2$$

where it can be assumed that $\zeta_3 = 0{\cdot}5$.

By introducing these expressions into equation (7), and combining with equation (6), a differential equation is obtained which can be integrated to yield :

$$c = c_o + \frac{g\,.\,H_{mo}}{L} \left[\frac{t}{\frac{M_o\, t}{\mathcal{J}\,\omega_o} + 1} \left\{ 1 - \frac{1}{H_{mo}} \left(0{\cdot}32\, c^2{}_{01} + \right. \right. \right.$$

$$\left. \left. \left. + \zeta_2 \frac{{c_0}^2{}_2}{2\,.\,g} + \frac{{c_0}^2{}_3}{2\,.\,g} \right) \right\} - \frac{H\,.\,t}{H_{mo}} \right]$$

When one pump is in operation, it is $c_{01} = 1{\cdot}27$ m. per sec. and $c_{02} = 4{\cdot}7$ m. per sec.; and when two pumps are operating it is $c_{01} = 2{\cdot}54$ m. per sec. The friction factor is a function of the valve opening as given in Fig. 4. By introducing numerical values, it is found

$$c = 1{\cdot}27 + 2{\cdot}02 \left[\frac{0{\cdot}88\, t}{0{\cdot}26\, t + 1} - 0{\cdot}96\, t \right] \quad .. \quad (9)$$

and if both pumps are operating, it is

$$c = 2{\cdot}54 + 2{\cdot}06 \left[\frac{0{\cdot}87\, t}{0{\cdot}26\, t + 1} - 0{\cdot}95\, t \right] \quad .. \quad (10)$$

For various times t the values for c are as follows :

	t =	0	1	2	3	
One pump	c_I =	1·27	0·73	−1·43	—	m. per sec.
Two pumps	c_{II} =	2·54	2·01	0·97	0·31	m. per sec.

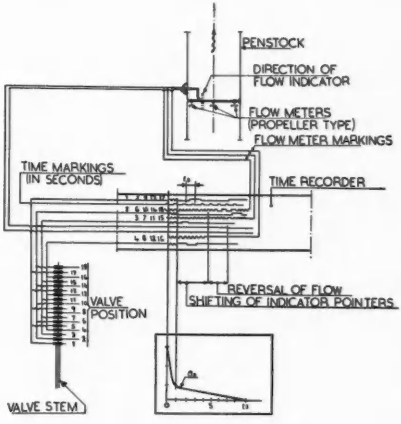


Fig. 6. Circuit of time recorder.

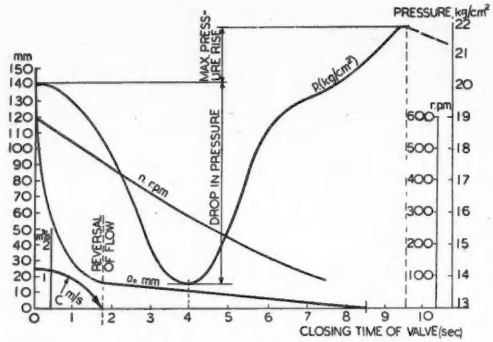


Fig. 7. **Test results found with shut-down of one pump showing water flow, change in water velocity, valve opening, and pressure variation as functions of valve closing time t.**

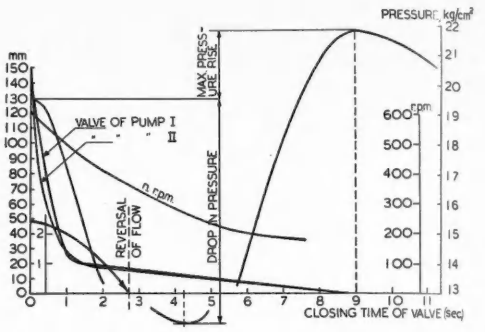


Fig. 8. **Same as Fig. 7 but with two pumps shut down.**

With one pump in operation, c attains zero after 1·75 sec., and in the case of two pumps being in operation, c becomes zero after 2·74 sec. The relationship between c and t is charted in Fig. 3.

In the tests carried out for the purpose of verifying the theoretical findings given above, a special flow direction indicator was installed in the penstock as shown in Fig. 5. This flow direction indicator is equipped with electrical contacts energizing the recording pen of a time recording instrument, the working principle of which is outlined in Fig. 6. The actual test results are charted in Figs. 7 and 8, where it will be seen that the water velocity attains zero after 1·75 sec. if one pump is operating (Fig. 7), and after 2·7 sec. if both pumps are in operation (Fig. 8). This result agrees well with the computed data. In fact, agreement is so close that it must be considered accidental, taking into account the various assumptions which had to be made with regard to the friction loss coefficient.

FRANCE

THE MANUFACTURE OF LUBRICATING OILS FROM ETHYL ALCOHOL.

(From *Le Génie Civil*, Vol. 123, No. 5, March 1, 1946, p. 64.)

ETHYL alcohol produced by the saccharification of beet root or of grain can serve as raw material for the production of lubricating oils comparable in quality to mineral oils and greases. The following two methods may be employed :

1. The production of olefines from ethyl alcohol and their subsequent dehydration, this to be followed by catalytical polymerization in the presence of aluminium chloride.

2. The production of ethylene from ethyl alcohol and its reaction with other hydrocarbons.

Production and polymerization of heavy olefines.

This is carried out by dehydrogenation of the alcohol to obtain acetaldehyde and subsequent condensation of the latter to form crotonaldehyde. From the latter butyl alcohol is then obtained by means of hydrogenation, thus :

$$\underset{\text{Ethyl Alcohol}}{2CH_3.CH_2OH} \longrightarrow \underset{\text{Butyl Alcohol}}{CH_3.CH_2.CH_2.CH_2OH} + \underset{\text{Water}}{H_2O}$$

Butylene is then obtained by dehydrogenation of the butyl alcohol, thus :

$$\underset{\text{Butyl Alcohol}}{CH_3.CH_2.CH_2.CH_2OH} \longrightarrow \underset{\text{Butylene}}{CH_2.CH:CH.CH_3} + \underset{\text{Water}}{H_2O}$$

Finally the butylene is polymerized. The reactions are not complete, as 100 kg of ethyl alcohol yield only 51·7 kg of butylene instead of the theoretical quantity of 61 kg. The polymerization of the butylene yields 47 kg of raw lubricating oil and a few pounds of drying oil which can be used for the preparation of paints and varnishes.

The polymerization is carried out by passing a stream of butylene into a vessel containing petroleum ether or a hydrocarbon of the type C_nH_{2n+2}. The bottom of the vessel is covered with a layer of finely pulverized aluminium chloride. By decanting, a clear liquid is obtained from which the aforementioned oils are produced by distillation.

Instead of preparing the aldehyde, one can also produce acetone and from this isopropyl alcohol by hydrogenation. The latter then yields propylene by dehydration. In the presence of aluminium chloride as catalyst the oil is then obtained by condensation. In this case—unlike the first process—no petroleum is obtained.

These oils are not very viscous but have satisfactory lubricating properties at all temperatures. They show little tendency to oxidation and thicken but little when oxidizing. In fact they are comparable or superior to a good grade of mineral oil and they show a very slow loss of lubricating properties in service.

The various fractions obtained by vacuum distillation can be used as transformer oil, turbine oil, and compressor oil respectively. These oils are fully miscible with mineral oil in all proportions.

In the ethylation process ethylnaphtalenes are prepared with the use of polyethylbenzenes. The latter are obtained by reacting ethylene with benzene in the presence of aluminium chloride as catalyst. This method of oil production is more expensive than the process carried out on the olefine basis, but these oils are likewise of good quality. Lubricating oils can also be produced by reacting ethylbenzene with dichlorethane in the presence of aluminium chloride as catalyst. These oils are of considerable viscosity and of high lubricating quality, but they show less resistance to oxidation than the aforementioned oils.

By means of these methods one ton of lubricating oil can easily be produced per hectar of beet root or Jerusalem artichoke.

CALCULATION OF THE PERFORMANCE CHARACTERISTICS OF PLAIN BEARINGS.

By A. FRÄNKEL. (From *Mitteilungen aus dem Institut für Thermodynamik und Verbrennungsmotorenbau, Eidgenössische Technische Hochschule Zürich*, No. 4, 1944, 134 pages, 65 illustrations.)

INTRODUCTION.

It is accepted practice to take a great deal of trouble in calculating the operating variables of bearings of infinite width to a high degree of accuracy. The data of the bearing of finite width are then obtained simply by multiplying these figures with a factor. This introduces an excessive error. The important rate of lateral oil flow out of the bearing cannot be estimated at all by this method. As a rule, a number of assumptions have to be made, and the methods of calculation are very cumbersome.

The author has developed a mathematical basis for calculating cylindrical plain bearings of finite width which permits estimating, with little expense of effort and time, the pressure distribution, resulting load, frictional torque, oil flow, etc., for any given bearing width and relative eccentricity. The approximate formulae are so simple in use that there is no difficulty in checking results or calculating a special case.

The second part of the study indicates methods of estimating data for bearings under periodically alternating loads. In the third part a simple way is shown for taking account of circumferential variations in viscosity.

NOMENCLATURE.

b axial bearing width

$\beta = \frac{b}{d}$ relative bearing width

r, d, R, D radius and diameter of journal and bearing

$\Delta r = R - r = \frac{D-d}{2}$ radial clearance

$\psi = \frac{\Delta r}{r} = \frac{\Delta d}{d}$ relative bearing clearance

e eccentricity

ϵ relative eccentricity

h film thickness

h_0 film thickness for $\varphi = \varphi_0$ when $\frac{\partial p}{\partial \varphi} = 0$

m exponent of parabola of axial pressure distribution

q_s, Q_s oil volume passing through one unit width, total oil volume

Q_R oil volume flowing out laterally at both bearing ends

Q_{tot} total oil volume passing through in x-direction

P, P_ξ, P_ζ resultant journal load and its components in ξ- and ζ-direction

$p, p_m = \frac{P}{bd}$, p_g local, mean and maximum pressure

p_0 local pressure in central plane of bearing

$\Pi = \frac{p\psi^2}{\eta\omega}$, $\Pi_m = \frac{p_m\psi^2}{\eta\omega}$, $\Pi_g = \frac{p_g\psi^2}{\eta\omega}$, $\Pi_0 = \frac{p_0\psi^2}{\eta\omega}$ dimensionless expressions for p, p_m, p_g, and p_0

S, S_R shear force acting on journal in loaded bearing half and unloaded half, respectively

$S_{tot} = S + S_R$ shear force acting on journal over whole circumference

$V_1, V_2 (= V)$ circumferential speed of bearing and journal

W frictional work (mKg./sec.) developed in loaded bearing half

Z frictional work per Kg. of oil passing through, mKg./Kg.

$z = \frac{Z\gamma\psi^2}{\eta_m\omega}$ dimensionless expression for Z

Z_{total}, z_{total}, total frictional work developed in loaded bearing half per Kg. oil

$x = r\varphi$ length of arc on journal periphery, measured from h_{max}

y co-ordinate in direction of film thickness, measured from bearing surface

z co-ordinate in axial direction, measured from central bearing plane

u, v, w velocity components in direction of x, y and z

$\varphi = \frac{x}{r}$ circumferential angle, measured from h_{max}

φ_0, φ_{p_g} angle for $\frac{\partial p}{\partial \varphi} = 0$

η dynamic viscosity, Kg.sec.m^{-2}

η_m mean viscosity

ω angular velocity of journal, sec^{-1}

λ, ν frequencies of rotation and oscillation

μ, μ_{total} coefficient of friction for loaded bearing half and for the whole journal perimeter respectively

ξ, ζ directions of co-ordinates

(A). BEARING OF FINITE WIDTH UNDER CONSTANT LOAD.

Basic Assumptions.—The following simplifying assumptions are made: the surfaces of journal and bearing are absolutely smooth; the axes of journal and bearing are exactly parallel; the clearance is finite, and small in comparison with the radius of the journal; inertia forces are negligible as compared with viscosity forces, the curvature of the clearance space is neglected; the velocity component in radial direction and the variation in pressure in radial direction are negligible; oil entirely fills the clearance space, adheres to the surfaces, is non-compressible and has a constant viscosity.

Fundamental Equations.—Based on Newton's equation for the shear stress between two liquid layers in parallel motion,

$$\tau = \eta \frac{\partial u}{\partial y} \quad .. \quad .. \quad (1)$$

the shear force per unit area of journal surface is calculated as

$$d S_{y=h} = dx\, dz\, \eta \left(\frac{\partial p}{\partial x} \frac{h}{2\eta} + \frac{V_2 - V_1}{h} \right) \quad .. \quad (2)$$

The oil volume passing through one unit of width in the X-direction (see Fig. 1) equals

$$q_s = h \frac{V_1 + V_2}{2} \quad .. \quad .. \quad (3)$$

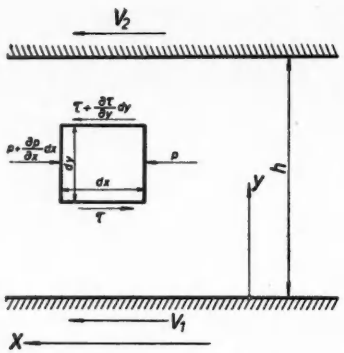


Fig. 1. Equilibrium of a volume element.

if pressure is constant in the direction of movement.

For the limiting surfaces at rest, the oil flow per unit width per second equals

$$q_{px} = -\frac{h^3}{12\,\eta}\,\frac{\partial p}{\partial x} \qquad .. \quad .. \quad (3a)$$

and in the z-direction

$$q_{pz} = -\frac{h^3}{12\,\eta}\,\frac{\partial p}{\partial z} \qquad .. \quad .. \quad (3b)$$

The general differential equation for pressure distribution in an oil wedge is then developed as :

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\,\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{h^3}{\eta}\,\frac{\partial p}{\partial z}\right) = 6\,(V_1 + V_2)\,\frac{dh}{dx} \quad (4)$$

This equation has also been developed by Reynolds, but so far only Vogelpohl has integrated it for a cylindrical bearing.

It is assumed that, in axial direction, pressure is a parabolic function, so that the pressure at any given point :

$$p = p_0\left[1 - \left(\frac{z}{b/2}\right)^m\right] \qquad .. \quad .. \quad (4a)$$

where p_0 is the pressure in the central plane. If equation (4a) is differentiated and introduced into equation (3b), after several transformations the following equation is obtained :

$$p_0\,\frac{h^3 m}{3\eta\,b} + b\,\frac{V_1 + V_2}{2}\,\frac{dh}{dx} - \frac{m}{m+1}\,\frac{b}{12}\,\frac{d}{dx}\left(\frac{h^3}{\eta}\,\frac{dp_0}{dx}\right) = 0 \quad (5)$$

which holds good for any given dependency of η and h upon x.

Equations for the Cylindrical Bearing.

For a cylindrical bearing, with a relative eccentricity

$$\epsilon = \frac{e}{\Delta r},$$

the clearance

$$h = r\psi\,(1 + \epsilon\cos\varphi) \qquad .. \quad .. \quad (6)$$

This equation and its derivation are introduced in equation (5). In order to simplify the resulting expression the following factors are substituted :

$$\Pi = \frac{p\,\psi^2}{\eta\,w}\;;\;\Pi_0 = \frac{p_0\,\psi^2}{\eta\,w} \qquad .. \quad .. \quad (7)$$

$$A = (m+1)\left(\frac{d}{b}\right)^2 = \frac{m+1}{\beta^2} \qquad .. \quad (8)$$

TABLE I. VALUES OF A AND B.

m	A = β=0·5	0·75	1·0	1·25	1·50	1·75	2·0	B
2·0	12·0	5·333	3·00	1·920	1·333	0·980	0·75	9·000
2·2	12·8	5·689	3·20	2·048	1·422	1·045	0·80	8·727
2·4	13·6	6·044	3,40	2·176	1·511	1·110	0·85	8.500
2·6	14·4	6·400	3·60	2·304	1·600	1·176	0·90	8·308
2·8	15·2	6·756	3·80	2·432	1·689	1·241	0·95	8·143
3·0	16·0	7·111	4·00	2·560	1·778	1·306	1·00	8·000

$$B = 6\,\frac{m+1}{m} \qquad .. \quad .. \quad .. \quad (9)$$

and the following basic formula results :

$$\frac{d^2\Pi_0}{d\varphi^2} - 3\,\frac{\epsilon\sin\varphi}{1+\epsilon\cos\varphi}\,\frac{d\Pi_0}{d\varphi} - A\Pi_0 = -B\,\frac{\epsilon\sin\varphi}{(1+\epsilon\cos\varphi)^3} \quad (10)$$

Values for A and B are shown in Table I.

Various pressure distribution patterns have been examined and as they have little effect on the resulting load, the pattern shown in fig. 2 has been chosen permitting considerable simplifications.

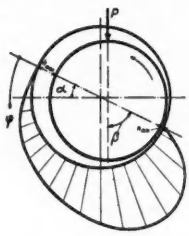


Fig. 2. Pressure distribution in bearing.

Transformation of the Equation.

Introducing a new variable

$$u = \Pi_0\,(1 + \epsilon\cos\varphi)^2 \qquad .. \quad .. \quad (11)$$

in an attempt to solve equation (10), by means of a Fourier-series

$$u = \sum_1^\infty C_n \,.\, \sin n\varphi \qquad .. \quad .. \quad (12)$$

Pressure Distribution.

Within the range of relative bearing width between $\beta = 0{\cdot}5$ and $\beta = 2{\cdot}0$ the third member of the Fourier series may be neglected in nearly all cases, causing, in general, less than 1 per cent error. A very simple equation for the pressure is found :

$$\Pi_0 = \frac{p_0\,\psi^2}{\eta\,\omega} = \frac{C_1\sin\varphi + C_2\sin 2\varphi}{(1+\epsilon\cos\varphi)^2} \qquad .. \quad (13)$$

in which

$$C_1 = \frac{B\,\epsilon}{1 + A + \frac{\epsilon^2}{2}\left(1 - \frac{A}{2}\right)} \qquad .. \quad (14)$$

and

$$C_2 = \frac{B\epsilon^2\left(1 - \frac{A}{2}\right)}{(4+A)\left[1 + A + \frac{\epsilon^2}{2}\left(1 - \frac{A}{2}\right)\right]} \quad (15)$$

TABLE II. VALUES OF C_1 FOR $m = 2{\cdot}4$.

ϵ	C_1 =								
	$b/d = 0{\cdot}5$	0·75	1·0	1·25	1·50	1·75	2·0	∞	1·304
0·0	0·000	0·000	0·000	0·000	0·000	0·000	0·000	0·000	0·000
0·1	0·0583	0·121	0·193	0·268	0·338	0·402	0·459	0·597	0·283
0·2	0·117	0·243	0·388	0·536	0·676	0·802	0·913	1·176	0·567
0·3	0·178	0·367	0·584	0,804	1·011	1·197	1·359	1·722	0·850
0·4	0·241	0·494	0·783	1·073	1·344	1·584	1·793	2·222	1·133
0·5	0·306	0·626	0·986	1·343	1·679	1·962	2·211	2·667	1·417
0·6	0·376	0·763	1·193	1·614	1·996	2·328	2·611	3·051	1·700
0·7	0·452	0·909	1·407	1·886	2·314	2·681	2·989	3·373	1·983
0·8	0·534	1·063	1·628	2·160	2·626	3·019	3·343	3·636	2·267
0·9	0·624	1·229	1·858	2·436	2·931	3·340	3·673	3·843	2·550
0·95	0·675	1·317	1·977	2·575	3·080	3·491	3·828	3·928	2·692
1·00	0·727	1·409	2·099	2·714	3·228	3·644	3·977	4·000	2·833

TABLE III. Values of C_2 for $m = 2{\cdot}4$.

ϵ	C_2 =							
	$b/d = 0{\cdot}5$	0·75	1·0	1·25	1·50	1·75	2·0	∞
0·0	0·00000	0·00000	0·00000	0·000000	0·00000	0·00000	0·00000	0·0000
0·1	−0·00192	−0·00234	−0·00183	−0·000382	0·00150	0·00350	0·00544	0·0149
0·2	−0·00774	−0·00940	−0·00733	−0·00153	0·00601	0·0140	0·0217	0·0588
0·3	−0·0176	−0·0213	−0·0166	−0·00344	0·0135	0·0313	0·0484	0·129
0·4	−0·0317	−0·0383	−0·0296	−0·00612	0·0239	0·0552	0·0851	0·222
0·5	−0·0505	−0·0606	−0·0466	−0·00956	0·0373	0·0854	0·131	0·333
0·6	−0·0744	−0·0887	−0·0677	−0·0138	0·0532	0·122	0·186	0·458
0·7	−0·104	−0·123	−0·0932	−0·0188	0·0720	0·163	0·248	0·590
0·8	−0·141	−0·165	−0·123	−0·0246	0·0934	0·210	0·317	0·727
0·9	−0·185	−0·214	−0·158	−0·0312	0·117	0·262	0·392	0·865
0·95	−0·211	−0·242	−0·178	−0·0349	0·130	0·289	0·431	0·933
1·00	−0·239	−0·273	−0·199	−0·0387	0·144	0·317	0·472	1·000

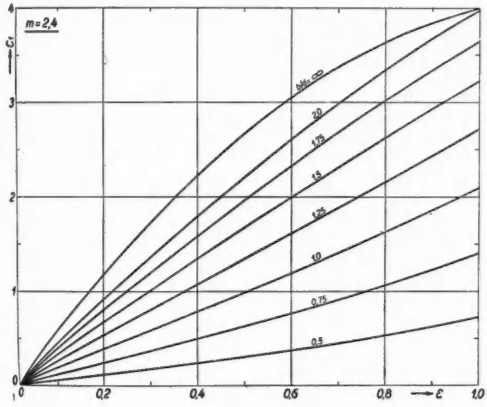


Fig. 3. **The coefficient C_1.**

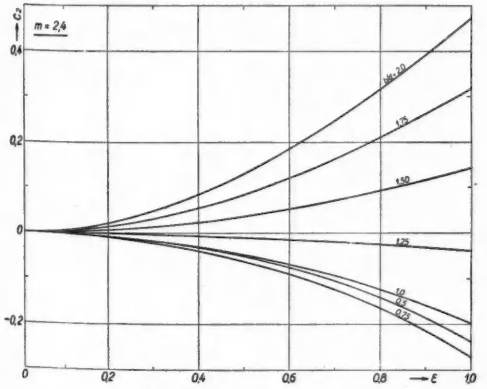
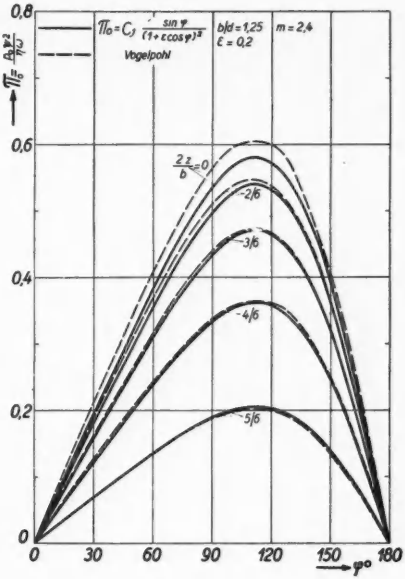
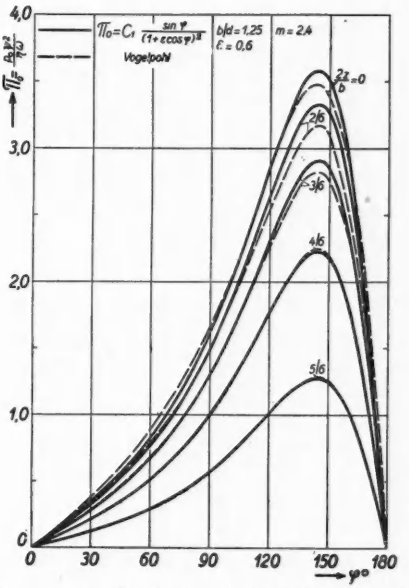


Fig. 4. **The coefficient C_2.**





Figs. 5-6. **Pressure distribution for b/d = 1·25 and $\epsilon = 0{\cdot}2$, $\epsilon = 0{\cdot}6$, in comparison with Vogelpohl's results.**

Factor C_1 is tabulated in Table II for different values of β and ϵ, and shown graphically in Fig. 3. Values of factor C_2 are given in Table III and Fig. 4.

For values of $\beta = b/d$ between 1 and 1·5 only small errors are caused if the second member of the Fourier series is neglected. Pressure distribution curves have been calculated using the simplified equation

$$\Pi = \frac{p\psi^2}{\eta\omega} = C_1 \frac{\sin\varphi}{(1 + \epsilon\cos\varphi)^2} \left[1 - \left(\frac{2z}{b}\right)^{2\cdot4}\right] \quad (16)$$

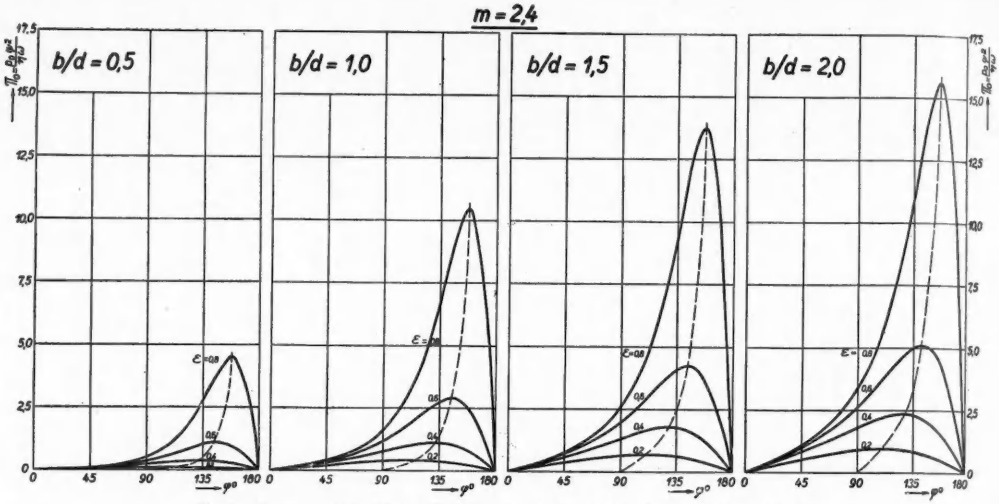


Fig. 7. **Pressure distribution in the central plane of bearings of finite width.**

for $\beta = 1{\cdot}25$, $m = 2{\cdot}4$ and $\epsilon = 0{\cdot}2$ and $0{\cdot}6$ for different values of $\frac{2z}{b}$ i.e. varying distances from the central plane.

Results have been plotted in Figs. 5 and 6. The dotted lines represent Vogelpohl's results and it is clear that agreement is excellent. The deviation nowhere exceeds 5 per cent. On the average it is much smaller. Consequently, equation (16) gives satisfactory results within the range $\beta = 1$—$1{\cdot}5$ for a small fraction of the work required by the method indicated by Vogelpohl.

Fig. 7 shows pressure distribution curves in the central plane calculated by equation (13) for different values of β and ϵ. Increase of bearing width above $\beta = 1.5$ causes no appreciable increase in the rate of pressure rise.

Resultant Journal Load.

The method developed for calculating the pressure distribution permits not only determining any given pressure distribution by the use of the simple formulæ developed but also calculating, by simple integration, the resultant journal load and the other operating factors of the bearing. Assuming for the unloaded bearing half between $\varphi = \pi$ and $\varphi = 2\pi$ a gauge pressure $p = 0$, based on Fig. 8.

$$\left.\begin{aligned} P_\zeta &= \int_0^{\pi}\int_{-b/2}^{+b/2} p\, r \sin\varphi \, d\varphi \, d\, z \\ P_\xi &= -\int_0^{\pi}\int_{-b/2}^{+b/2} p\, r \cos\varphi \, d\varphi \, d\, z \end{aligned}\right\} \qquad (17)$$

Transformations and introduction of equation (13) results in :

$$\frac{P_\zeta \psi^2}{\eta\, \omega\, b\, d} = \Pi_{m\zeta} = \frac{m}{2(m+1)}\left[C_1 \int_0^{\pi} \frac{\sin^2\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi + 2C_2 \int_0^{\pi} \frac{\sin^2\varphi\cos\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi\right] \quad .. \quad .. \quad (18)$$

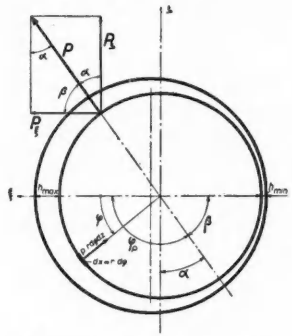


Fig. 8. **Calculation of the resultant journal load. (Symbols used.)**

$$\frac{P_\xi \psi^2}{\eta\, \omega\, b\, d} = \Pi_{m\xi} = -\frac{m}{2(m+1)}\left[C_1 \int_0^{\pi} \frac{\sin\varphi\cos\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi + 2C_2 \int_0^{\pi} \frac{\sin\varphi\cos^2\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi\right] \quad .. \quad .. \quad (19)$$

The integrals can be calculated as follows :

$$\int_0^{\pi} \frac{\sin\varphi\cos\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi = \frac{1}{\epsilon^2}\ln\frac{1+\epsilon}{1-\epsilon} - \frac{2}{\epsilon}\,\frac{1}{1-\epsilon^2} \qquad (20)$$

$$\int_0^{\pi} \frac{\sin^2\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi = \frac{\pi}{\epsilon^2}\left(\frac{1}{\sqrt{1-\epsilon^2}} - 1\right) \qquad (21)$$

$$\int_0^{\pi} \frac{\sin^2\varphi\cos\varphi}{(1+\epsilon\cos\varphi)^2} d\varphi = \frac{\pi}{\epsilon\sqrt{1-\epsilon^2}}\left(1 - \frac{2}{\epsilon^2}\right) + \frac{2\pi}{\epsilon^3} \qquad (22)$$

TABLE IV. Values of integrals from equations (20)–(23).

ϵ	$\int_0^\pi \frac{\sin\varphi\cos\varphi}{(1+\epsilon\cos\varphi)^2}d\varphi$	$\int_0^\pi \frac{\sin^2\varphi}{(1+\epsilon\cos\varphi)^2}d\varphi$	$\int_0^\pi \frac{\sin^2\varphi\cos\varphi}{(1+\epsilon\cos\varphi)^2}d\varphi$	$\int_0^\pi \frac{\sin\varphi\cos^2\varphi}{(1+\epsilon\cos\varphi)^2}d\varphi$
0·0	0·0000	1·5708	0·0000	0·6667
0·1	−0·1350	1·5827	−0·07933	0·6793
0·2	−0·2800	1·6195	−0·1636	0·7171
0·3	−0·4478	1·6855	−0·2588	0·7876
0·4	−0·6568	1·7885	−0·3733	0·9029
0·5	−0·9389	1·9440	−0·5209	1·0889
0·6	−1·3575	2·1817	−0·7272	1·4001
0·7	−2·0622	2·5664	−1·0480	1·9705
0·8	−3·5113	3·2725	−1·6362	3·2226
0·9	−8·0608	5·0194	−3·1461	7·3866
0·95	−17·5331	7·6671	−5·5506	16·3990
1·00	$-\infty$	∞	$-\infty$	∞

$$\int_0^\pi \frac{\sin\varphi\cos^2\varphi}{(1+\epsilon\cos\varphi)^2}d\varphi = \frac{2}{\epsilon^2}\left(1+\frac{1}{1-\epsilon^2}\right) - \frac{2}{\epsilon^3}\ln\frac{1+\epsilon}{1-\epsilon} \quad .. \quad .. \quad .. \quad (23)$$

These integrals have been calculated for a number of values of ϵ and tabulated in Table IV. This, together with the values for C_1 and C_2 given in Tables II and III, allows calculating, for $m = 2{\cdot}4$, the components of the resultant load for any given bearing width and relative eccentricity.

TABLE V. Values of $\Pi_m = \dfrac{p_m\,\psi^2}{\eta\,\omega}$ for $m = 2{\cdot}4$

ϵ	$b/d = \beta =$ 0·5	0·75	1·0	1·25	1·5	1·75	2·0	∞
0·1	0·0329	0·068	0·109	0·150	0·190	0·225	0·257	0·472
0·2	0·0697	0·143	0·226	0·311	0·391	0·463	0·526	0·951
0·3	0·115	0·233	0·365	0·496	0·618	0·727	0·822	1·446
0·4	0·177	0·351	0·540	0·724	0·892	1·040	1·167	1·976
0·5	0·268	0·518	0·782	1·029	1·254	1·439	1·599	2·577
0·6	0·415	0·779	1·148	1·477	1·759	1·996	2·192	3·319
0·7	0·679	1·236	1·771	2·219	2·585	2·881	3·117	4·373
0·8	1·253	2·208	3·062	3·720	4·221	4·601	4·891	6·244
0·9	3·134	5·320	7·118	8·339	9·167	9·731	10·120	11·438
0·95	7·139	11·846	15·551	17·814	19·221	20·067	20·608	21·535

The total load is determined (see Fig. 8) as

$$P = \sqrt{P_\xi^2 + P_\zeta^2}, \quad .. \quad .. \quad (24)$$

or dimensionless :

$$\frac{P\psi^2}{\eta\,\omega\,b\,d} = \frac{p_m\cdot\psi^2}{\eta\,\omega} = \Pi_m = \sqrt{\left(\frac{P_\xi\psi^2}{\eta\,\omega\,b\,d}\right)^2 + \left(\frac{P_\zeta\psi^2}{\eta\,\omega\,b\,d}\right)^2} \quad .. \quad (25)$$

in which p_m is the average load per unit area. Determining Π_m as a function of m, β and ϵ leads to a very complicated equation and the indicated method is much easier.

Table V gives $\Pi_m = \dfrac{p_m\,\psi^2}{\eta\,\omega}$ as a function of ϵ for

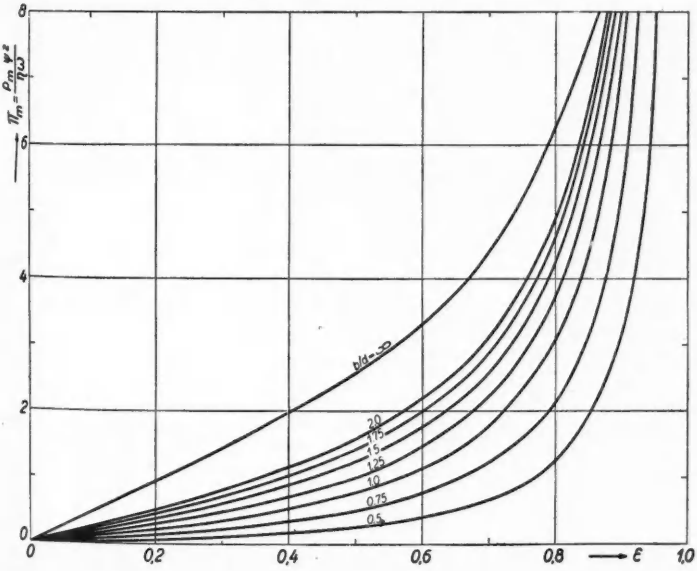


Fig. 9. **Resultant journal load for any given bearing width.**

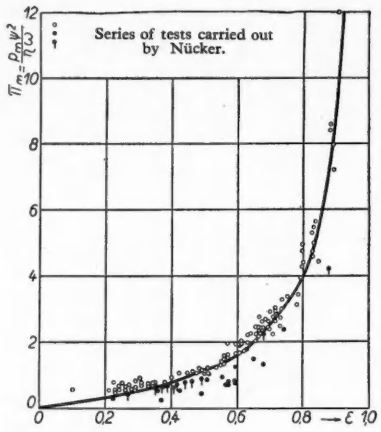


Fig. 10. **Resultant journal load for b/d = 1·36 compared with test results by Nücker.**

different values of β. The figures for $b/d = 1{\cdot}25$ are almost identical with those given by Vogelpohl, except for very high relative eccentricities. For $b/d = \infty$ agreement is complete with figures calculated by Gümbel. The values from Table V have been plotted in Fig. 9 which permits, for any given relative bearing width, to establish the correlation between resulting load and relative eccentricity.

These results have been compared with the very reliable test results given by Nücker (Fig. 10). The relative bearing width $\beta = 1{\cdot}36$. Agreement is very good and scatter is chiefly caused by variations in viscosity and by the difficulty of choosing the right mean value for η. The full circles are for a clearance ψ of half the value of that for the other points. Fig. 9 shows that, for a given ϵ, the load carrying capacity of a bearing greatly depends on β, and that for a given mean load Π_m, the position of the journal in the bearing, i.e. ϵ, is also greatly dependent of β. Hence, it is concluded that a correct evaluation of the various factors in a bearing of finite width can only be attained if the bearing with infinite width is abandoned as a mathematical basis and the calculation is carried out directly for a given bearing width.

(*To be continued*).

RUSSIA

HIGH TEMPERATURE METALLURGICAL FURNACE WITH INJECTOR-TYPE BURNERS FOR PREHEATED AIR AND GAS OF LOW HEATING VALUE.

By K. M. Holzman and B. R. Imenitov. (From *Stal*, No. 4, 1945, pp. 124-126. 3 illustrations).

In high temperature reheating furnaces excellent results are obtained with injector-type burners in which the combustion air is introduced into the gas stream. In this connection the use of a gas of low calorific value is of special interest. The main advantages inherent in the use of injection type burners are the following :

1. The gas is burned with a minimum of incomplete combustion.
2. The combustion of the gas is effected with a minimum of excess air.
3. High rates of heat release in the combustion space are effected, the rate of heat release being as high as 1,500,000 kal per cu.m. per hour and higher.
4. Fans and air ducts are dispensed with.
5. High thermal economy.
6. The metal in the furnace can be heated to high temperatures, up to 1200-1250 deg. C., in spite of the employment of a gas of low calorific value.
7. High specific furnace floor ratings are obtained, amounting to as much as 500-600 kg. per sq.m. per hour, even where the charge has to be heated from cold to 1200 deg. C.

The chief disadvantages of the injection type burner are :

1. Variations in the fuel/air ratio if the air or gas preheat changes, entailing re-adjustment of the excess air ratio.
2. The requirement of gas pressures ranging from 350-500 mm. w.g.

In Table I is shown the increase in furnace temperature (Δt_n) as function of the air preheat (t_a). The furnace temperature t_n prevailing if no air or gas preheat is used, is also shown.

Table I.

t_n deg. C.	t_a =	100	200	300	400	500	600
1000	Δt_n =	29	57	86	114	143	172
1100	Δt_n =	32	63	94	126	157	189
1200	Δt_n =	34	69	103	137	171	206

The subjoined Table II shows the increase in the furnace temperature (Δt_n) with the gas preheat (t_r) based upon a gas of a calorific value of 1300 kal per cu.m. N.P.T. The table also shows the furnace temperatures prevailing under identical conditions if no gas or air preheat is used.

Table II.

t_n deg. C.	t_r = 100	200	300	400	500	600
1000	24	48	72	96	120	144
1100	26	53	79	106	132	158
1200	29	58	86	115	144	173

The increase in thermal economy—expressed in per cent of reduction in gas consumption for a gas of the aforementioned calorific value—effected by the use of air or gas preheat is as follows :

Table III.

Influence of Air Preheat.

Air preheat, deg. C.	0	100	200	300	400	500
Reduction in gas consumption, per cent	0	11	21	28	34	40

Influence of Gas Preheat.

Gas preheat, deg. C.	0	100	200	300	400	500	600
Reduction in gas consumption per cent	0	10	18	24	30	35	39

These figures have been confirmed by tests conducted on actual furnaces. Where a gas of lower calorific value than given above is used or where higher metal tem-

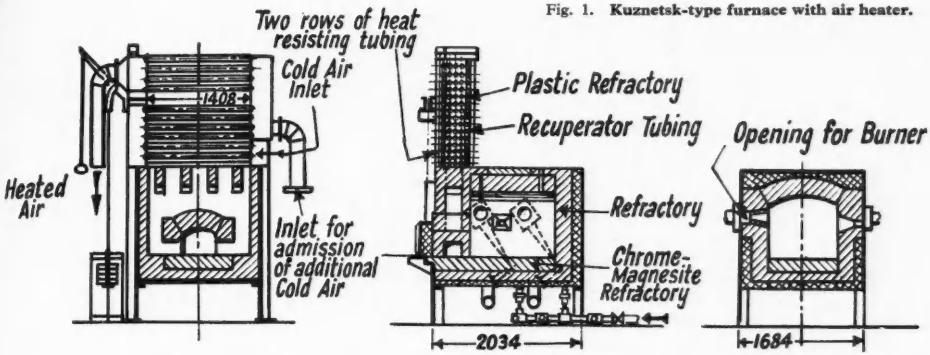


Fig. 1. **Kuznetsk-type furnace with air heater.**

peratures are required, or where the furnace output is higher, the savings achieved by gas or air preheat are also correspondingly higher.

The extent to which the air or the gas may be preheated is limited by :

1. The velocity of discharge of the gas-air mixture from the burner outlet. This velocity must be higher than the speed of flame propagation in the mixture.
2. The temperature of self-ignition of the mixture. This approximates to 500-580 deg. C.
3. The possibility of ignition of the gas by contact with hot metal at the point in the burner where mixing of gas and air takes place.

The last two points are of particular importance in cases where high preheats are employed, such as air temperatures of 800-900 deg. C. But such temperatures cannot be obtained in recuperators in which waste furnace gas is used as heating medium. The first factor mentioned is therefore by far the most important and is the decisive one in practice.

It may be assumed that the speed of flame propagation increases in proportion with the square of the absolute temperature of the gas-air mixture, or, what is the same, that the speed of flame propagation based upon a temperature of 0 deg. C. is increased in proportion with the absolute temperature of the mixture.

AIR HEATERS.

The velocity of the air in the recuperator must be 3-5 m. per sec., based upon N.P.T., and the total friction loss coefficient of recuperator and air supply duct must be 0·2-0·3 times the dynamic pressure equivalent of the gas-air mixture in the aspirating part of the burner. This approximates to a pressure loss in recuperator and air supply duct equivalent to 1·5-3·0 per cent of the gas pressure prevailing on the up-stream side of the aspirator; this corresponds to a gas pressure drop of 15-30 per cent.

GAS HEATERS.

The permissible flow resistance in the gas heater may be 30 to 300 mm. w.g. and more, and it may correspond to a drop in gas pressure of 10-30 per cent based on the gas pressure prevailing ahead of the aspirating part of the burner. This goes to show that a greater flow resistance can be allowed for in the gas heater than in the air heater, a gas velocity of 10-30 m. per sec. in the gas heater being permissible. Because of the high gas velocity employed in the gas heater, the temperature drop in this apparatus is some 2-3 times greater than in the air heater. Consequently, the space requirements of the gas heater are 1·5-2 times smaller than those of the air heater.

Fig. 1 shows a furnace of the Kuznetsk type with a floor area of 1·1 sq.m. used for the heating of steel stock from 0 to 1200 deg. C. This furnace is seen to be equipped with an air heater, four injector type burners being provided. The design of the burner and its dimensions are given in Fig. 2. The tube bundle of the air heater is composed of 2 ins. tubing with the tube rows arranged in alignment. The air is drawn into the heater by the aspirating effect of the injectors, a gas pressure of 350 mm. w.g. being required. The hot-air ducts of the heater are covered with heat insulating material. The temperature of the gas-air mixture is 150-160 deg. C. Further design data are given below :

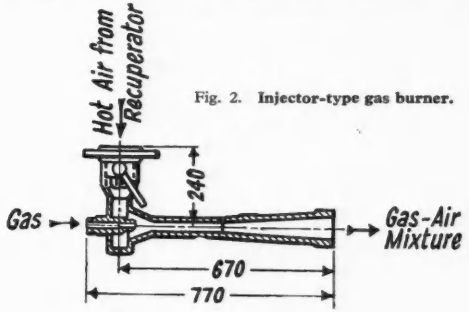


Fig. 2. **Injector-type gas burner.**

Furnace floor, sq.m. 1·1 ;
Useful furnace volume, cu.m. 0·95 ;
Heating surface of recuperator, sq.m. 10 ;
Heat transfer coefficient of the recuperator, kal. per sq.m. hour deg. C. : 8 ;
Temperature rise of furnace charge, deg. C. from 0 to 1200 ;
Furnace capacity, kg. per hour, 280 ;
Specific capacity per sq.m. floor area, kg. per sq.m. hour : 255 ;
Fuel : Clean producer gas from Chelyabinsk coal ;
Calorific value of fuel, kal per cu.m. N.P.T. : 1200-1400 ;
Gas supply pressure, mm. w.g. : 1600 ;
Excess air factor : 1·0.

Air temperature without and with recuperator	0		300	
Calorific value of gas kal per cu.m. N.P.T.	1200	1400	1200	1400
Estimated combustion temperature, deg. C.	1620	1700	1740	1820
Fuel Consumption.				
'000 kal per hour	750	520	430	350
cu.m N.P.T. per hour	580	370	360	250
Specific heat consumption kal/kg.	2500	1850	1550	1250
Heat saving effected by air preheat, per cent	—	—	38	32
Minimum permissible gas pressure at nozzle of injector, mm. w.g.	120	120	500	500
Frictional resistance of recuperator mm.w.g.	—	—	40	40

U.S.A.

THE OXYACETYLENE FLAME.

By G. V. SLOTTMAN, Manager, Technical Sales Division, Air Reduction Sales Co., New York, N.Y.

BECAUSE of its large endothermal heat, liberated by thermal decomposition, a flame temperature of 6300 deg. F can be produced from the hydro-carbon gas acetylene (C_2H_2). This remarkably high flame temperature is obtained by burning the acetylene in oxygen at the orifice of a torch, at low ratios of oxygen to acetylene, performed in two stages of combustion. The primary combustion produces the high temperature heat which is most effective for welding, generated by the decomposition of acetylene and by the formation of carbon monoxide. During this primary stage, which occurs within a small bluish-white cone close to the torch, the hydrogen content remains unchanged since water vapour is largely dissociated at such high temperatures and low oxygen concentrations.

The secondary combustion burns the carbon monoxide to carbon dioxide and the hydrogen to water vapour, giving rise to the outer flame of low luminosity. The total heat generated by the complete combustion is made up as follows :—

HEAT PRODUCED BY THE COMPLETE COMBUSTION OF ACETYLENE.

Name of Constituent	Reaction Formula	B.Th.U. per Mol.	B.Th.U. per cu. ft. 60 deg. F. 1 Atm.	Per cent of Total
Carbon ..	$2C+2O_2=2CO_2$	352,000	928	64·8
Hydrogen	$H_2+\frac{1}{2}O_2=H_2O$	104,100	277	19·3
Endothermic	—	86,600	228	15·9
Total Heat of Combustion		542,700	1433	100·0

The three commonly used types of oxyacetylene flame, neutral, carburizing and oxidizing, are the results of burning acetylene with varying oxygen-acetylene ratios. When pure oxygen and acetylene are premixed, before burning, in equal volumes the neutral flame is produced by the primary combustion in the following reactions :—

PRIMARY COMBUSTION OF ACETYLENE IN NEUTRAL FLAME INNER CONE.

$$C_2H_2+O_2=2CO+H_2+195{,}200 \text{ B.Th.U.}$$

Thus, at a ratio of one-to-one the primary combustion of the neutral oxyacetylene flame results in the formation of carbon monoxide and hydrogen. 287 B.Th.U. per cu. ft. of acetylene are liberated by the combustion of carbon to form two molecular volumes of carbon monoxide. In addition, 228 B.Th.U. per cu. ft. of acetylene are liberated by the dissociation of acetylene, giving a heat output of 515 B.Th.U. (or 35·9 per cent of the total available heat) per cu. ft. of acetylene in the primary combustion zone. The term " neutral flame " is derived from the fact that neither of the two products of the primary combustion has any tendency to oxidize molten metal

SECONDARY COMBUSTION OF ACETYLENE IN OUTER OR ENVELOPE FLAME.

(*a*) Combustion of Carbon Monoxide :

$$2CO+O_2=2CO_2+243{,}400 \text{ B.Th.U.}$$

(*b*) Combustion of Hydrogen :

$$H_2+\tfrac{1}{2}O_2=H_2O+104{,}100 \text{ B.Th.U.}$$

Here, the burning of hydrogen to water vapour liberates 277 B.Th.U. per cu. ft. of acetylene. The combination of the two volumes of carbon monoxide provides an additional 641 B.Th.U. per cu. ft. for the reaction, or a total of 918 B.Th.U. per cu. ft. of acetylene. Taking place in a voluminous lower temperature flame surrounding the welding operation, the secondary combustion contributes a preheating effect which assists the high temperature inner flame cone.

By adding the B.Th.U. generated by each of the two reactions, we arrive at the total heat of combustion for each volume of acetylene : 1433 B.Th.U. per cu. ft. of the gas. The large portion of the heat (35·9 per cent) which is developed in the high temperature inner flame cone explains the great difference in combustion characteristics between acetylene and all other commercial heating fuels. The total heat developed by a neutral oxyacetylene flame can be more easily broken up and explained in the following equations :—

HEAT PRODUCED BY THE NEUTRAL OXYACETYLENE FLAME.

Name of Constituent	Reaction Formula	B.Th.U. per Mol.	B.Th.U. per cu. ft. 60 deg. F. 1 Atm	Per cent of Total
(1. Primary combustion—inner flame cone.)				
Carbon ..	$2C+O_2=2CO$	108,600	287	20·0
Endothermic	—	86,600	228	15·9
Total primary		195,200	515	35·9
(2. Secondary combustion—outer flame.)				
Carbon Monoxide	$2CO+O_2=2CO_2$	243,400	641	44·8
Hydrogen	$H_2+\frac{1}{2}O_2=H_2O$	104,100	277	19·3
Total secondary		347,500	918	64·1
Total primary and secondary combustion		542,700	1433	100·0

The one volume of oxygen employed to burn one volume of acetylene in the primary combustion must be of high purity in order to obtain the maximum temperature. The one and one-half volumes of oxygen required in the secondary combustion comes from the atmosphere by diffusion into the envelope flame and since it is accompanied by some four times as much nitrogen, the temperature of the secondary or envelope flame is substantially lower than that of the primary flame. Although the commonly considered complete oxidation stage is reached with a ratio of two-and-one-half-ton-to-one, the dissociation of CO_2 and H_2O at these high temperatures results in the maximum flame temperature being obtained with oxyacetylene ratios in the region of 1·7 to 1. The flame temperature slowly decreases from that point with increasing oxygen additions although the total heat liberated continues to increase.

A carburizing flame, consisting of molecular and atomic hydrogen, carbon monoxide, and incandescent carbon particles resulting from the thermal decomposition of acetylene, is produced by oxyacetylene ratios below one to one. This flame retains the sharply defined inner cone and bluish outer envelope of the neutral flame, with a third recognisable zone of whitish colour between them, made up of particles of solid carbon.

Fusion welding of high melting point metals can only be accomplished by the tip of the high temperature inner flame cone. In cutting metals, also, the inner cones of the preheating flames quickly raise the surface to kindling temperature. Hence, the efficiency of any fuel gas for welding or cutting is primarily determined by its ability to liberate the maximum percentage of its total heat at the highest possible flame temperature in the inner cone. No other fuel gas liberates as great a percentage of its total heat value at such extremely high temperatures as does acetylene.

With increasing ratios above one to one the flame becomes increasingly oxidizing, with a short, purple-tinged inner cone, some carbon monoxide and hydrogen burning to carbon dioxide and water vapour, with free

oxygen appearing in the flame. Because of the interdependence of the flame temperature and the degree of dissociation of the combustion products, the exact calculation of the flame temperature is an involved problem and has no direct solution. At temperatures above 3000 deg. F. carbon and hydrogen do not completely combine with oxygen to form CO_2 and H_2O. At temperatures above 4000 deg. F. the hydrogen molecule itself becomes increasingly unstable, dissociating into hydrogen atoms. The degree of this dissociation increases rapidly with temperature and with a reduction in the partial pressure of the flame constituents, which depends on the percentage concentration of the combustion products and on the total flame pressure. At 6000 deg. F., 80 per cent of the CO_2 and 50 per cent of the H_2O are dissociated. Dissociation limits maximum attainable oxyacetylene flame temperatures which would otherwise go much higher. The proportions of the products of primary combustion in the inner cone depend upon the amount of oxygen present in the premixed gases leaving the torch tip. The following table shows these products, the heat liberated, and the flame temperature of the inner cone, with various mixing ratios of oxygen to acetylene.

PRIMARY COMBUSTION OF ACETYLENE IN INNER FLAME CONE.‡

Type Flame	Mol. Vol. of Combustion Products						Heat Liberated *		Flame Temp. Deg. F.†
	C§	CO	CO_2	H_2	H_2O	O_2	B.Th.U.	Per cent of Total	
arb.	·4	1·60	0	1·00	0	0	173,000	31·7	5,550
arb.	·2	1·80	0	1·00	0	0	184,000	33·9	5,700
n. ..	0	2·00	0	1·00	0	0	195,000	35·0	5,850
. ..	0	1·80	·20	·70	·30	·25	250,000	46·1	6,200
. ..	0	1·72	·28	·56	·44	·61	275,000	50·7	6,100
. ..	0	1·65	·35	·51	·49	1·10	288,000	53·2	6,000

* Total heat liberated for complete combustion of acetylene, 542,700 B.Th.U. per Mol. Vol.

† Theoretical. Actual flame temps. are somewhat lower because of radiation.

‡ Based on 1 molecular volume of acetylene = 380 cu. ft. at 60 deg. F. and Atm.

§ 4 molecular weights of solid carbon = 4·8 pounds.

It can be seen that neutral and carburizing inner flame cones are of somewhat lower temperature, and liberate less heat than oxidizing cones. However, since acetylene liberates its endothermic heat by thermal decomposition even without combustion, a strongly carburizing flame holds up remarkably well from both viewpoints.

To rate fuel gases only on the basis of total heat evolved during combustion, however, is erroneous, since the proper classification is also dependent upon the volume of the products of combustion, a factor which would show acetylene first with 97·0 B.Th.U. per cu. ft. of air combustion products. This heat intensity rating shows butane, for instance, to be third with only 84·4 B.Th.U. per cu. ft.; whereas, by rating only on the basis of total heat generated during combustion, butane shows 3107 B.Th.U. per cu. ft. of butane while acetylene develops only 1433 B.Th.U. per cu. ft. of acetylene.

Economy in the heating of metals for welding and cutting is governed by two factors: the temperature of the oxy-fuel gas flame—which determines the speed of operation—and the "heat availability," or that portion of the total heat of combustion of the fuel gas which is available above the working temperature of the process. Using steel as an example, where the weld face and the weld rod must be melted, sufficient heat must be generated above 2800 deg. F. by fuel gas combustion to cause the weld metal to flow freely. On a strictly thermal basis the heat unit of acetylene is more valuable than the heat unit of other fuel gases, because a higher percentage of it is available for use at elevated temperatures. To illustrate this principle, it is a known fact that cold producer gas and air will not develop a sufficient supply of heat at the melting point of steel, regardless of the total heat supplied to the furnace, to support the open hearth process, and that both the producer gas and air must be heated before the flame temperature is raised above the working temperature of the process.

U. S. A.

MECHANICAL ADVANCEMENT IN THE JOINING OF METALS AND OTHER MATERIALS.

By GEORGE H. TINNERMAN, Vice-President, Tinnerman Products, Inc.

THE SPRING TENSION SPEED NUT SYSTEM.

Described by engineers as one of the biggest manufacturing advances since the advent of the continuous assembly line, the spring tension speed nut system has truly revolutionized the assembly of hundreds of mass production products. The speed nut principle of joining metals and other materials was originally applied to help the stove industry solve its complicated assembly problems. But within a few short months, the idea had expanded and found ready use in the manufacture of any product where reduced costs and stepped-up production are "must" factors. Automobiles, refrigerators, washers, furnaces, radios, toys and countless other items were assembled better and faster through using the speed nut system. Even before Pearl Harbour, speed nuts were ordered at the rate of more than 3,000,000 per day. War time manufacture ran into billions, and it is probable that similar figures will continue as the production industries reconvert.

THE SPEED NUT SYSTEM IS DIFFERENT.

The principle of the speed nut differs radically from previous methods of joining two or more pieces of material. No other fastening device provides a thread lock as well as a self-energizing spring lock. In the starting position (Fig. 1) the two arched prongs of the nut fit into the threads of the screw, while the base of the nut is still well arched. As the screw is turned and tightened, the arched prongs move inward to engage and lock against the root of the thread. These free-acting prongs compensate for tolerance variations, and function equally well on both under and oversize threads. And a second and equally important lock develops as the screw is tightened. A self-energizing spring lock is created by the combined compression of the arched prongs and the base. Thus, two distinct forces are exerted as the screw is tightened; the compensating thread lock and the self-energizing spring lock, a combination which eliminates all possibility of the nut loosening from vibration. Speed nuts bite harder and tighter into the threads of the screw when the load is increased. This is only one of the many basic features of the speed nut idea which sets it apart from all other fasteners.

ADVANTAGES OF THE SYSTEM.

Many unique advantages are gained through using the speed nut system of assembly for metals, plastic, porcelain, glass and other materials.

1. Being made of spring steel, the nuts are under constant spring tension and will not loosen under the

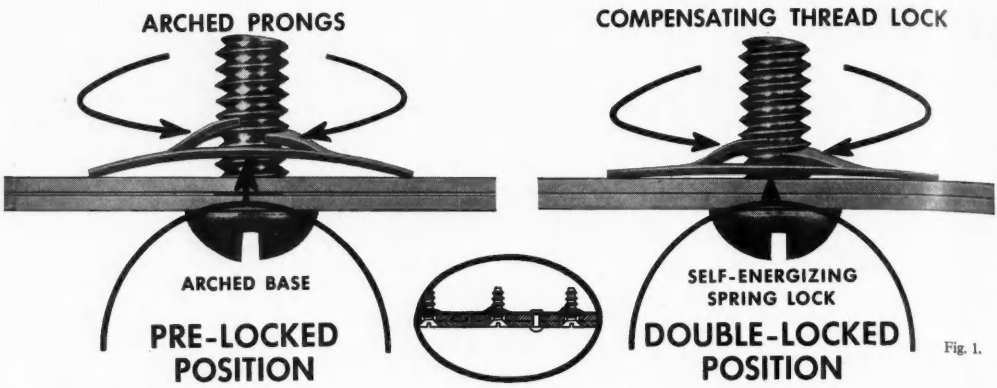


Fig. 1.

most severe vibration. Aircraft vibration tests of 4000 strokes per minute found speed nuts holding tight after hundreds of hours of this severe punishment.

2. The free acting prongs of the nuts compensate for tolerance variations, permit a certain amount of " float " and they will lock on over or undersize screw threads.

3. Being lighter than ordinary nuts, they effect a weight reduction of 70 to 80 per cent.

4. Where old style, rigid nuts invariably cracked enamel, scratched plastic and broke glass, speed nuts, with their resilient spring tension can not damage such surfaces.

5. Having no threads, the free acting speed nut prongs can not clog with paint or other foreign matter.

6. Production lines report savings of from 50 to 80 per cent in assembly time after switching from threaded nuts to spring tension nuts which catch onto screw threads instantly.

7. Spring tension nuts combine nut and lock washer into one unit.

8. Some types of speed nuts are self-retaining, holding themselves in bolt-receiving position. Thus welding, riveting or clinching operations are eliminated.

APPLICATIONS OF THE SPEED NUT PRINCIPLE.

The basic spring tension speed nut, a single unit stamped from a coil of spring steel (see Fig. 1), rapidly assumed new shapes as new uses were found for it. The " twin " type came next, and then the multiple nuts in strips, made with any required spacing and in any length (see Fig. 2). There are some 3000 different applications of the speed nut idea and new uses, demanding new designs, are almost a daily occurrence. Some of the more common types are worthy of comment.

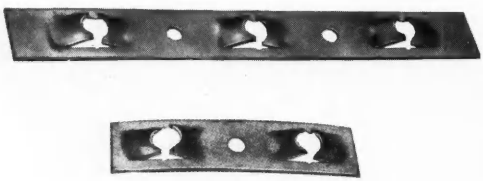


Fig. 2.

A SPEED NUT DESIGN FOR EVERY USE.

The progressive nature of the speed nut principle soon permitted the production of the " U " and " J " types of nuts which hold themselves in blind positions when the holes are close to the edge of the material. This eliminates the need for riveting or welding nut plates at those locations (see Fig. 3). A series of speed nut angles combines an angle bracket and fastener in one piece, reducing the number of parts usually needed for such installations (see Fig. 4). The speed nuts specially made for plywood have two pairs of integrally formed attaching legs. When driven into thick plywood, the cam-like structure of the legs forces them outward, permanently holding the speed nuts with a firm spring tension grip in the wood. In thin plywood, the legs " peen " over against a backing plate (see Fig. 5).

Push-on type speed nuts are applied to rivets or unthreaded metal or plastic studs with a simple tool which presses them securely into a locked position (see Fig. 6). Another fastening for unthreaded studs or rivets in blind location assembly is the tubular type of speed clip. These are snapped into holes where attachment is possible from one side only. After a second part of assembly is lined up, a rivet or stud is then pressed into the clip, causing the clip to expand and grip the rivet, making possible a fully concealed fastener that grips securely and is installed instantly. Both permanent locking and removable types are available (see Fig. 7).

A " hi-stress " speed nut meeting Army Air Force specifications No. 25531 is extremely light in weight, and has unusually low installation torque which permits rapid insertion of bolts or screws. Its self-locking torque is retained after many removals under service conditions (see Fig. 8).

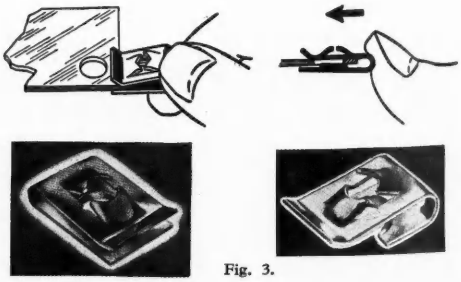


Fig. 3.

Fig. 4

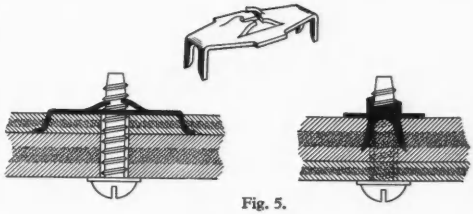


Fig. 5.

Fig. 6

Fig. 7

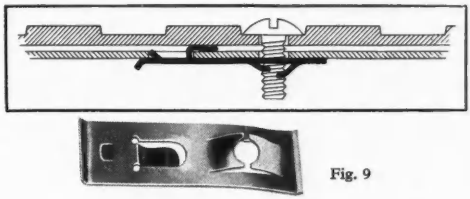


Fig. 9

Fig. 8

Latching type speed nuts snap into round or rectangular holes in mid sections of panels or sheets, where the holes are too far from the edge to permit the use of the " U " or " J " type. Latching nuts eliminate the costly riveting or welding of nut plates and save assembly time (see Fig. 9). The speed nut cover plate was designed for patching bullet holes in the metal skins of airplanes, but soon it was being used as a removable cover for inspection doors, access holes and vent holes. They are quickly attachable from one side by sliding one end of the speed nut into the hole, centering the cover plate and tightening the screw (see Fig. 10).

OTHER ITEMS IN THE SPEED NUT SYSTEM OF ASSEMBLY.

Included in the spring tension system for reducing assembly costs are various latches, clips and clamps, all made from spring steel and developed in line with the basic speed nut principle. For example, to meet the demand for a simple, secure, vibration-proof means of holding any removable assembly, a spring tension latch was designed. The spring arms of the speed latch firmly grip the ball studs or grooved studs, depending on the type of assembly to be inserted. This latch is used on access doors, glove compartments, box covers, handhole plates, etc. (see Fig. 11). A recently developed clip provides a speedy, efficient means of attaching the covers of metal, plastic or wooden boxes. These clips, two or more to the box, are applied entirely outside the box with nothing inside to obstruct wires or other equipment. Clips are snapped into position by hand or with a simple tool (see Fig. 12). The box cover is locked under spring tension and attachment and removal require no special tools.

A hose clamp, all in one piece because the locking element is an integral part of the clamp itself, with no gears, screws or other loose members, recently became a feature of the speed nut system. To install, the clamp is snapped into prelatched position by hand, and then locked securely with one-hand compression of standard 6 in. pliers. Removal is equally simple (see Fig. 13). Cable clips have been developed to a high degree of perfection. They are equally applicable to automobiles, radios, ranges, boats and wherever cables are used. Cable clips are hand snapped into round holes in panels or sheets, eliminating bolts, nuts, washers, special brackets or clamps. The cable is then snapped into the spring arms of the clips for enduring vibration-proof, spring tension assembly (see Fig. 14).

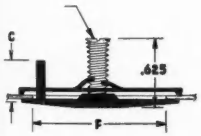


Fig. 10.

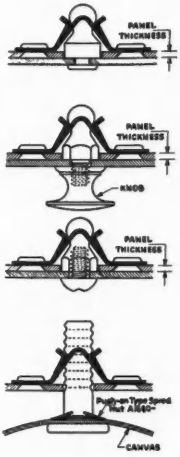


Fig. 11.

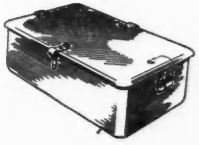


Fig. 12.

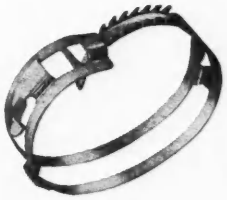


Fig. 13.

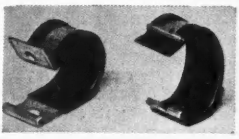


Fig. 14.

OPTICAL METHOD FOR THE DETERMINATION OF THE SURFACE OF POWDERS IN SUSPENSION.

By H. BAUMANN. (From *Teknisk Tidskrift*, Vol. 75, No. 32, August 11th, 1945, pp. 893-895, 5 illustrations).

THE physical and technical properties of certain plastic materials depend to a high degree upon the particle size of the material. Materials of this kind are pulverized in ball mills; the grinding action was investigated in detail by means of the optical method described below.

It is a well known fact that in the wet grinding of black powder, as for instance, lamp black or graphite, the suspension of the powder in water becomes darker with increasing length of the grinding. This is easily explained by the fact that the light can only pass through the unobstructed areas of the suspension. But these areas decrease with increasing dispersion of the powder in the liquid.

If it is assumed that the particles in suspension are larger than the wave length of the light, then 1 gram of powder will obturate a certain amount of light comparable to the light absorption of a homogeneous solution.

If $\overline{P}$ is the obturation effected by 1 gram of powder, dx is the thickness of the suspension passed by the parallel beam of light rays, c in grams per cu. cm. is the concentration of the suspension, and $\mathcal{J}$ is the light intensity, then

$$-d\mathcal{J} = \mathcal{J}\,\overline{P}\,c\,dx$$

$$\frac{\mathcal{J}}{\mathcal{J}_o} = e^{-\overline{P}cx}$$

$$\overline{P} = \frac{\ln \mathcal{J}o - \ln \mathcal{J}}{cx} = \frac{2{\cdot}30\,(\log \mathcal{J}o - \log \mathcal{J})}{cx} \quad .. \quad (1)$$

Here $\overline{P}$ corresponds to the absorption coefficient of a homogeneous solution and constitutes a measure of the degree to which the powder is dispersed in the liquid. In the case of black powders, the particle size of which is not less than 1 micron, the absorption coefficient can be measured with ordinary light and with the use of an absorption tester of the usual type, as, for instance, a Pulfrich photometer. Coloured powders require the employment of a complementary light in order to avoid errors of measurement due to reflection.

The relationship between the obturation $\overline{P}$ and the particle size can be found as follows: Assuming that the powder consists of spherical particles of identical radius r and identical specific gravity s, the number of particles per gram of substance will be:

$$n = \frac{1}{4\,\pi\,r^3 s/3} = \frac{3}{4\,\pi\,r^3 s}$$

and therefore

$$\overline{P} = n\,\pi\,r^2 = \frac{3\,\pi\,r^2}{4\,\pi\,r^3 s} = \frac{3}{4\,r\,s} \quad .. \quad (2)$$

However, the assumption of identical particle size will rarely hold true in practice; and the physical meaning of the value for $\overline{P}$ obtained from equation (2) is therefore highly problematical. In view of the proportionality between $\overline{P}$ and r, it is obvious that the value for $\overline{P}$ will not apply to some mean particle radius, but will be mainly determined by the radius of the smallest particles present.

In many cases, however, the physical-technical properties of a powder are better characterized by its total surface than by its particle size. For spherical particles, irrespective of their size, the ratio of light absorption to the powder surface A will be given by

$$\frac{\overline{P}}{A} = \frac{\Sigma\,\pi\,r^2}{\Sigma 4\,\pi\,r^2} = \frac{1}{4} \quad .. \quad .. \quad (3)$$

It can be shown that equation (3) does not only hold good for spherical particles, but quite often for any particles having convex surfaces. In the latter case it will again be:

$$\overline{P} = \Sigma\,\Delta\,\overline{P} = \Sigma\,\Delta\,\frac{A}{4} = \frac{A}{4} \quad .. \quad (4)$$

By combining equations (1) and (4) the surface of the powder is found as:

$$O = 9{\cdot}2\,\frac{(\log \mathcal{J}_o - \log \mathcal{J})}{c\,.\,x} \quad .. \quad .. \quad (5)$$

If the particles are not convex, the ratio $\frac{\overline{P}}{A}$ will be smaller than $\frac{1}{4}$.

It may be asked what practical value, if any, can be attached to equation (5), considering that it is impossible to ascertain whether or not the individual particles are convex. This question cannot be answered to the full, but it can at least be said that the possible error incurred is without any practical importance. Actually, it is not the surface of the particles as such which is of interest, but such properties as the power of adsorption, coating power, etc. In the case of a pencil, for instance, the strength and wear of the lead depend largely upon the degree of dispersion of the graphite. Similar considerations apply to certain rubber products in which lamp black is added to the rubber. Moreover, the influence of the particle surface upon the specific properties is not always the same. Fig. 1 shows a convex particle in which the surface has a very large number of fissures, similar to that of active carbon or of diatomaceous earth with extremely high power of adsorption. It is obviously impossible to determine the active surface of particles of this kind by means of formula (4). If particles of the type shown in Fig. 1 are contained in the lead of a pencil, the large number of fissures will have a most beneficial effect upon the strength and wearing qualities of the product. The clay particles L in Fig. 1 are too large to be able to enter into the fissures. Even if this should happen, it would be highly improbable to have any influence, as the mechanical properties of the mixture obviously depend upon the peripheral surface of the particle, which can be determined from formula (4) or, if its surface is convex, from formula (5).

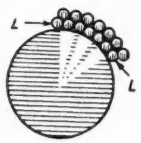


Fig. 1

CHARGING OF STORAGE BATTERIES.

By F. DACOS. (From *Revue Universelle des Mines*, Series 9, Vol. II, No. 1, 1946, pp. 3-22. 34 illustrations.)

INTRODUCTION

THIS article is based on the results of a large number of experiments on the conditions of charge and discharge of storage batteries. Transient conditions, although of theoretical interest, are not dealt with, as these affect only a small fraction (about 5 per cent.) of the total energy involved and are consequently of no interest to the consumer.

The first part of the article deals with the general behaviour of both lead and alkaline storage batteries, which can be simply explained by the phenomenon of diffusion. In addition to theoretical explanations, a large number of experimental data are given. The second part, which is thought to be of considerable practical interest, is abstracted in the following : Its principal conclusions are that the total charging current of unequally discharged batteries of the same type which are directly connected to a common supply is distributed between the batteries automatically in the correct way without the use of regulating resistances; the latter, though recommended by other authors can only have adverse effects and should never be used ; that if any number of batteries are connected in parallel for charging, only one amperemeter and one voltmeter are required for control and that both can be rigidly connected ; finally, that the life of batteries charged with pulsating current exceeds by about 30 per cent that of batteries charged with smooth D.C.

THEORETICAL AND EXPERIMENTAL ANALYSIS OF THE CHARGE

1. *Lead Cell.*—Charge and discharge of a battery are reversible processes. In the first case, however, the SO_4 ions of the electrolyte, which penetrate into the active material of the positive plate to transform the PbO_2 into $PbSO_4$, move about inside the layer of the plate under the influence of both the diffusion and the electrostatic attraction.

The density of the SO_4 ions in the layer where the sulphatation of the lead peroxide, i.e. the chemical reaction which liberates the electric charges, takes place obviously depends on the thickness of the sulphate layer the ions have had to penetrate. This fact has been taken into account in the investigations into the discharge of a lead cell.

On the other hand, during the charge, the number of ions produced per second on the surface of contact of the PbO_2 and the $PbSO_4$ depends on the intensity of the charging current, i.e. on the capacity of the charging set ; the density of the ions in the layer is, therefore, a function of the current, which means of the voltage V applied to the terminals. The ions formed at the surface of contact of the PbO_2 and the $PbSO_4$ cross the lead sulphate and increase the density of SO_4 ions in the electrolyte. This increase varies from place to place but, given sufficient time for diffusion to take place, the density will become uniform. The intensity i of the current is given by Ohm's law : $i = \frac{V - e}{R}$, where e is the e.m.f. of the cell (opposed to V) and R the resistance of the circuit.

Experience shows that the e.m.f. of a cell is an almost exactly linear function of the density of the ions at the place where the chemical reaction takes place.

It can be assumed that, before the commencement of the charge, sufficient time has elapsed for the diffusion to bring about a uniform distribution of the density δ_e of the electrolyte at all points of the cell, including the liquid itself as well as the volume of the plate.

The charging of the cell increases the density δ_1 at the surface of contact of the PbO_2 and the $PbSO_4$, and it can be said that the intensity of the charging current is proportional to the number of SO_4 ions regenerated per second, and therefore to the difference $\delta_1 - \delta_e$ of the densities.

When charging is in progress, more and more layers become involved and the density δ_1 increases with time ; consequently, the current intensity diminishes if the voltage at the terminals is kept constant.

It follows that, in considering a normally discharged cell of a traction battery, the final density of the electrolyte is $\delta_e = 18$ deg. Baumé ; if the cell is three-quarters discharged, the density is of the order of 22 deg. Baumé, and if it is half discharged the density is of the order of 25 deg. Baumé.

The ion density corresponding to an e.m.f. of 2·4 volts is about 32 deg. Baumé (which corresponds to a fully-charged battery). Or, if the potential difference at the terminals of a normally charged cell is 2·4 volts, the density at the surface of the active material can be assumed to be 32 deg. Baumé.

The charging current at the commencement of the charge is proportional to $(\delta_1 - \delta_e)$, i.e. to $32 - 18 = 14$. If an equal voltage of 2·4 volts is applied to the terminals of a cell three-quarters discharged, the difference of density is $32 - 22 = 10$, and the charging current will be proportional to 10. For a cell one half discharged, the charging current is proportional to $32 - 25 = 7$.

This means that, all other conditions being equal, the charging current is proportional to the degree of discharge of the cell.

Summarizing the above, it can be said that, at any moment during the charging, the e.m.f. E of a cell is equal to the e.m.f. E_0 of the discharged cell plus an e.m.f. proportional to the amount of electricity Q already fed

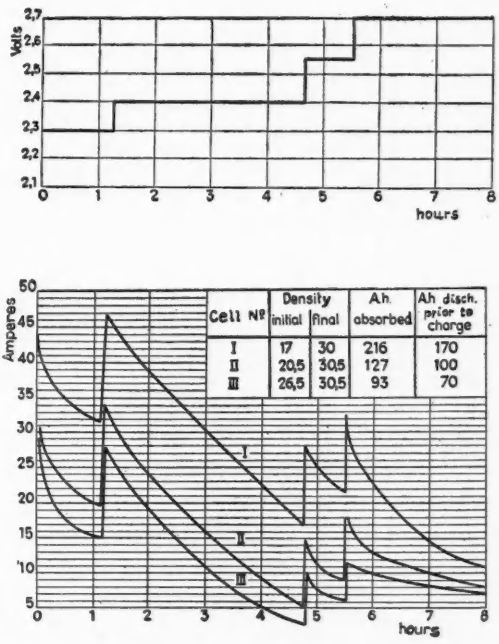


Fig. 1. **Charging voltage and current of three lead cells connected in parallel as a function of time.**

into the cell ; or, if K is the corresponding factor, $E=E_0+KQ$.

From this it follows that, if a number of unequally discharged cells in parallel are connected to a source of electricity of 2·4 volts, the total charging current is distributed between the cells in such a way that the charge of all the individual cells will be completed at exactly the same time.

This theory has been confirmed conclusively by a large number of experiments, the results of one of these experiments are shown in Fig. 1. The three batteries were connected in parallel without resistances to the terminals of the charging set. The stepping up of the charging voltage was done in arbitrary stages.

2. *Alkaline cells.*—Although in the case of alkaline cells th density of the electrolyte in the cell does not vary in the course of charge or discharge, experiments show that there are considerable fluctuations of the density in the electrodes, which means that there is a complete analogy between alkaline and lead cells. This has been borne out by experiments, the results of which show a marked similarity with the results obtained for lead cells (Fig. 2). However, in the case of alkaline

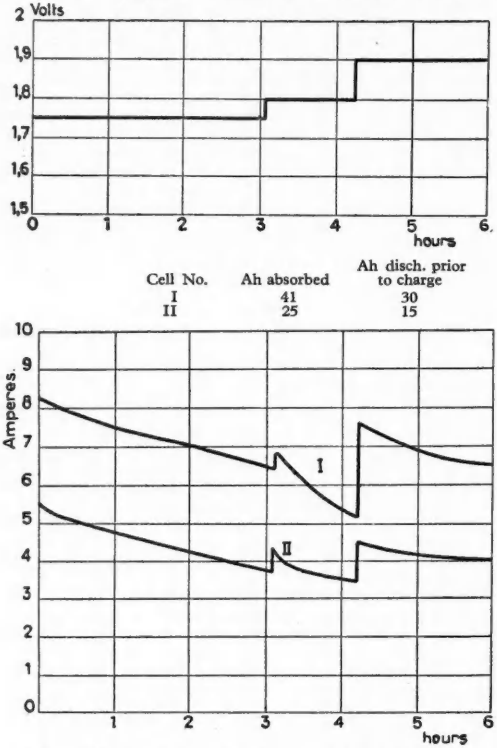


Fig. 2. **Charging voltage and current of Two Ni-Fe cells connected in parallel as a function of time.**

cells, the charging current should always be of the order indicated by the designer : if the intensity is excessive the proportional relationship between the current and the discharge in ampere-hours will not be maintained ; if, on the other hand, the charging current is too small, the cells discharged to a lesser degree will not take on the charge, which is a well-known property of Ni-Fe cells.

3. *Low temperature tests.*—Fig. 3 shows the measured Ah-efficiencies of a lead cell of large capacity and of a Ni-

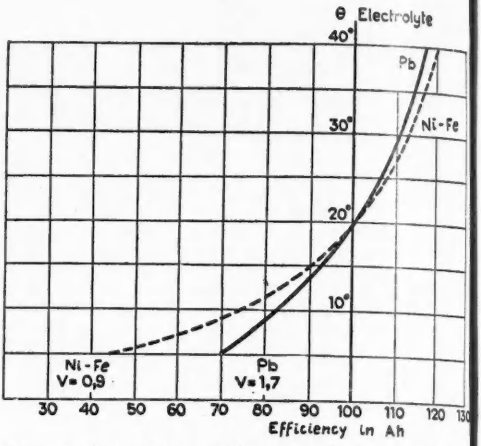


Fig. 3. **Variation of Ah efficiency of a lead and a Ni-Fe cell as a function of the temperature during the discharge. (The efficiency at 20 deg. C being considered normal).**

Fe cell of small capacity at temperatures between 5 deg. C and 40 deg. C.

Although both curves cannot be directly compared, they both clearly indicate a considerable decrease of the Ah-efficiency at lower temperatures, a phenomenon which can be explained by the slower rate of diffusion at these temperatures.

4. *The charge of lead cells.*—It is often required, especially in connection with electric traction, to charge simultaneously a large number of batteries ; in such cases it is imperative that the charging should be done correctly, with the smallest possible expenditure of apparatus, and in the most economic way. A general practical solution of the problem thus set was found by the following theoretical investigations into the behaviour of two cells I and II connected in parallel to the bus-bars of a source of electricity (Fig. 4). Since there are no limitations as to the number of cells, the results obtained can be extrapolated for any required number of cells.

The following assumptions are made :—

1. The voltage V at the terminals of cell I is kept constant.
2. The resistances ρ of the leads connecting the cells to the bus-bars are equal, and the internal resistances of both cells (included in ρ)—are constant.
3. The circuit of cell II contains an additional resistance R (for instance the resistance of the bus-bars between cells I and II ; a possible difference between the internal resistance of the cells could be added to R).
4. The variations of the e.m.f. follow the equation $E=E_0+KQ$ as previously explained. E_0 is the e.m.f. of the discharged cell.

(a) Assuming first that the cells are discharged to the same degree, so that

$$Q_0^{I}=Q_0^{II}$$

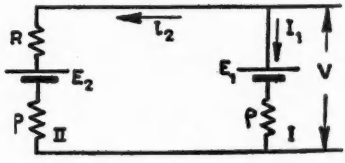


Fig. 4.

Ohm's law applied to cell I gives :

$$I_1 = \frac{V - E_1}{\rho}, \qquad \text{wherefrom}$$

$$\rho \frac{dI_1}{dt} = -\frac{dE_1}{dt}; \quad \frac{dE_1}{dt} = K\frac{dQ_1}{dt} = KI_1$$

$$\frac{dI_1}{I} = -\frac{K}{\rho}dt, \text{ and } I_1 = Ce^{-\frac{K}{\rho}.t}$$

For $t=O$: $\quad I_1 = I_{10} = \frac{V-E_0}{\rho}$

$$I_1 = \frac{V-E_0}{\rho} \cdot e^{-\frac{K}{\rho}.t} \qquad .. \quad .. \quad \text{(I)}$$

and $$Q_1 = \frac{V-E_0}{K}\left[1-e^{-\frac{K}{\rho}.t}\right]$$

If E_f is the final voltage, Q_f the total amount of electricity accumulated, namely $Q_f = Q_0 - Q_1$, it follows from $E = E_0 + KQ$:

$$K = \frac{E_{f1}-E_0}{Q_1}$$

Ohm's law applied to cell II gives :

$$I_2 = \frac{V - I_2R - E_2}{\rho}$$

$$(\rho + R)\frac{dI_2}{dt} = -\frac{dE_2}{dt} = -KI_2$$

$$I_2 = \frac{V-E_0}{\rho+R} \cdot e^{-\frac{K}{\rho+R}.t} \qquad .. \quad .. \quad \text{(II)}$$

$$Q_2 = \frac{V-E_0}{K}\left[1-e^{-\frac{K}{\rho+R}.t}\right]$$

$$K = \frac{E_{f2}-E_0}{Q_2}$$

This value of K is the same as for cell I, cells I and II being similar.

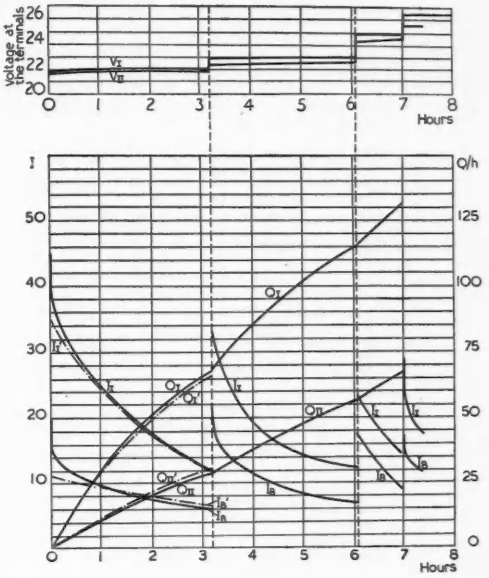


Fig. 5. Charging voltage and current of two lead cells.

(b) The same calculation can be applied where the degree of discharge is not the same for both cells ; however, the correct values for E_0 for the given state of discharge of each cell have to be introduced.

The validity of the above calculation has been confirmed by experiments. One example is shown in Fig. 5, where the dotted curves represent the calculated and the fully drawn curves the measured values. The two cells under investigation were of the same type ; cell I was completely discharged (172 Ah), cell II only half discharged (86 Ah) ; the e.m.f. of cell I was 1·97 volts, and that of cell II 2·1 volts.

It has also been found that the small voltage drop in the bus-bars from which a large number of batteries are charged is of no practical consequence.

(To be continued.)

SWITZERLAND

THE PLUNGER-COIL VOLTAGE REGULATOR.

By A. MEYERHANS. (From *The Brown Boveri Review*, Vol. 32, No. 6, June, 1945, pp. 207–210, 7 illustrations).

VARIOUS systems are being used for the regulation of the secondary voltage at points of consumption. Of these, regulators having no contacts are to be preferred, particularly for out-of-the-way sites. This category, to which belong the conventional induction regulator and the sliding coil transformer, has been enriched by a new type, the plunger-coil regulator. While, in the case of the former two systems, one wound iron core is moved in relation to another, in the plunger type regulator only a winding is moved in relation to a core and yoke system carrying an energizing winding (Fig. 1). In the end position A the coil W_T is inside the core and linked with the entire magnetic flux produced by coil W_E ; thus the maximum voltage is induced in coil W_T. In position B, however, coil W_T is not linked with the magnetic flux and no voltage is induced. Between these two end positions the coil moves in an annular gap, the voltage induced in the coil varying according to the particular position. As a rule, the plunger and energizing coils are auto-transformer-connected, since only a limited range of regulation (e.g., 20 per cent) is usually required. However, the plunger coil can be connected to the energizing winding also in boost or buck or through the intermediary of a small auxiliary winding. For special applications, the plunger and energizing windings can be made independent for regulation from zero.

The radial core has a high upper yoke with the result that the flux density in the gap is only a small fraction of that in the core and that the regulator does not draw an excessive no-load current despite the rather wide annular gap.

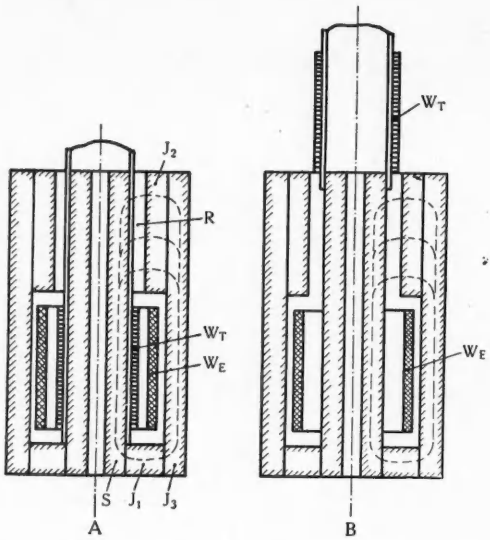


Fig. 1. **Fundamental design of a single-phase plunger-coil voltage regulator.**

The coil W_T moves in and out of magnetic linkage.

A End position, inserted	R Annular gap
B End position, withdrawn	S Radially laminated core
J_1 Lower yoke	W_E Energizing coil
J_2 Upper yoke	W_T Plunger coil
J_3 Return magnetic circuit	

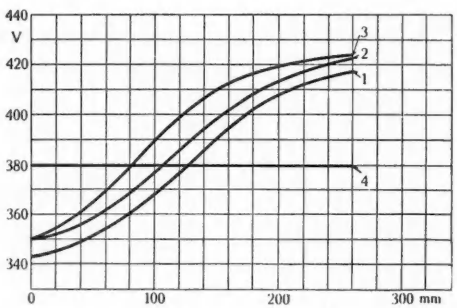


Fig. 4. **Relation between unregulated voltage and path of regulating coil in a plunger-coil voltage regulator for a 25 A through-current.**

Regulated voltage = 380 V. (Buck connection with auxiliary winding.)

(1) Unregulated voltage at no-load.

(2) Unregulated voltage at 25 A. load and p.f. = 1.

(3) Unregulated voltage at 25 A. load and p.f. = 0·7.

(4) Regulated voltage (380 v.).

Abscissæ :—Path of regulating coil.

Ordinates :—Unregulated and regulated voltages.

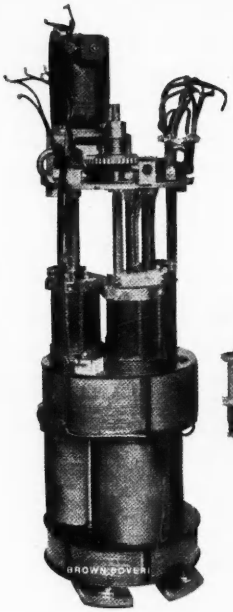


Fig. 2. **Active parts of a three-phase plunger-coil voltage regulator for a 50 A through-current.**

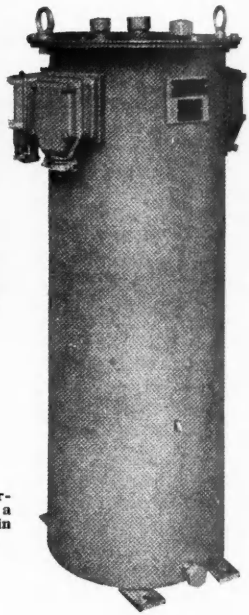


Fig. 3. **Three-phase plunger-coil voltage regulator for a 50 A through-current in weather-proof oil-tank.**

For three-phase operation the three cores are preferably arranged in triangular formation and fitted in round yoke plates (Fig. 2). The plunger coils are wound on bakelized paper cylinders attached to a spider which moves up and down a threaded spindle like a travelling nut. The spindle is located in the centre of triangularly arranged cores and is rotated by a small motor over a reduction gear. The connexion to the plunger coil is made by very flexible copper cables. The whole, including the control motor, is housed in a cylindrical tank filled with oil, which makes the transformer suitable for outdoor installation; the smaller types can be mounted on poles. In the latter case, the control relay, housed in a waterproof box, is conveniently mounted low down the pole; this enables the transformer to be readily set for the required regulated voltage. The control relay can be adjusted to a sensitiveness of $\pm 0{\cdot}2$ to $\pm$ 3 per cent. If a small current transformer and a compounding resistance are installed it is possible to regulate the secondary voltage in relation to the current, instead of to a constant value.

Over the whole regulating range the voltage varies sufficiently uniformly for practical purposes.

ELECTRICAL MEASUREMENT OF FORCES IN PRESS TOOLS.

By F. FAULHABER. (From *V.D.I. Zeitschrift*, Vol. 88, No. 25/26, June 24, 1944, pp. 347–352, 22 illustrations).

SPECIAL types of measuring devices for press tools had to be developed in consequence of the rough treatment of such tools and high pressures involved. The results obtained with these, however, gave valuable information on the various press tool processes and are described hereafter. The experience gained with such special devices can be utilized to develop and improve existing press tool processes.

When measuring forces of heavy drawing or injection presses, forging machines and the like, the forces involved are very much greater than those encountered in other fields of engineering.

Some hydraulic presses are provided with hydraulic safety cushions and an indicating instrument can be connected to these. Such measurements, however, do not give fully reliable information, particularly regarding transient pressures. The purpose of this paper is to discuss the measurement by electrical means of forces with devices which can easily be incorporated in standard presses.

There are various methods of inserting a measuring device into a press. In the case of small embossing presses and the like the measuring head can be placed underneath the tool (see Fig. 1). For heavy drawing presses and most cold injection processes it is preferable to insert the measuring device into the plunger (see Fig. 2).

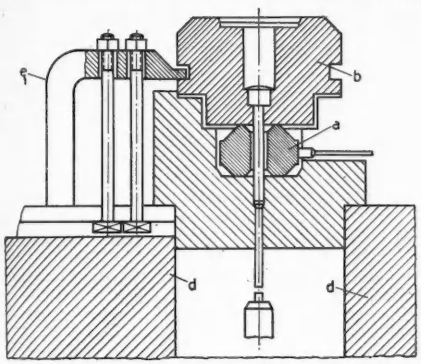
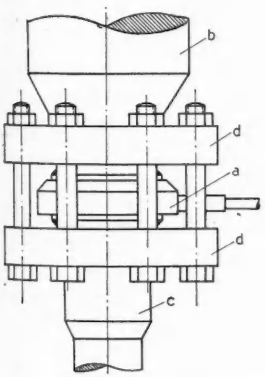


Fig. 1. **Assembly of a measuring head underneath the press tool.**
(*a*) Measuring head; (*b*) Tool; (*c*) Ejector; (*d*) Press table; (*e*) Holder.



(*a*) Measuring head.
(*b*) Plunger of the press.
(*c*) Drawing plunger.
(*d*) Clamping device for the measuring head.

Fig. 2. **Assembly of a measuring head between plunger and drawing tool.**

In this case it is possible to pre-stress the measuring device in order to be able to measure forces in the withdrawing cycle.

Pressure measuring devices for substantial forces are usually made in such a manner that the force to be measured induces an elastic longitudinal deformation of the body of the measuring device. In consequence of the high pressures the devices have to be rather rigid and the elongations are therefore essentially small.

PRACTICAL REQUIREMENTS REGARDING PRESSURE MEASURING DEVICES

The following conditions have to be laid down for measuring devices :—

(1) Low weight, smallness, suitable shape and capacity for taking large overloads.

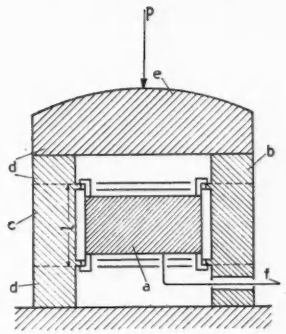
(2) Independence of eccentric loading.

(3) High ceiling of frequency, insensitivity against factors like high temperature, humidity, etc.

(4) Linear calibration curve and equal slope of the curves of individual heads (this is necessary in order to be able to use several heads on one press).

SHAPE OF MEASURING HEAD.

Most of the measuring heads are designed in such a manner that the actual measuring device is located inside a hollow resilient body according to Fig. 3. This



(*a*) Measuring device fixed to gauge length *l*.
(*b*) Resilient cylinder of the cross section *q*.
(*c*) Active material of the head.
(*d*) Inactive material of the head.
(*e*) Percussion cap.
(*f*) Connecting lead.

Fig. 3. **Schematic diagram of a measuring head with internal measuring device.**

design provides good protection against harmful external influences. In order to avoid making the connecting meter too sensitive it is preferable to make the elongation Δl rather large, for instance between ·01 and ·1 mm. According to Hooke's Law $\Delta l = Pl/qE$ where l is the gauge length, P the force, q the cross section and E the modulus of elasticity. If a definite value of Δl is taken and a definite compression force $\sigma = P/q$ is used, the gauge length l can be determined. The weight of the internal part of the resilient body is $G = l.q.\gamma$ where γ is the specific gravity. Combining the above equations the weight of the active part of the resilient body follows $G = \Delta l P E \frac{\gamma}{\sigma^2}$

A good comparison for the weight of individual measuring heads can be found in the specific load, i.e., that load which can be carried by a measuring head per kg. of its own weight :

$$L = P/G = \sigma^2/\Delta l E \gamma$$

For a constant Δl, i.e., constant sensitivity, the weight decreases with the square of the specific load. For hydraulic measuring devices L = 1–3 tons/kg., for electrical devices approximately 2–10 tons/kg. The

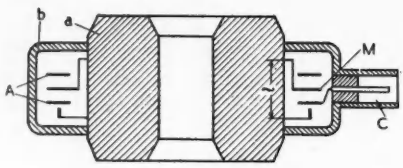


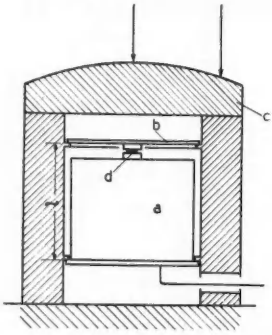
Fig. 4. **Measuring head with external measuring device (differential condenser).**

a) Resilient body ; (*b*) Case ; (*c*) Cable connexion. (*A*) External condenser plates fixed to the lower end of *l* ; (*M*) Intermediate condenser plate fixed to the top end of *l*.

head, according to Fig. 4, has the remarkable specific load of 77 tons/kg.

MEASURING DEVICES UNDER ECCENTRIC LOADING.

It frequently happens that press tools are elastically deformed themselves and therefore load the measuring head eccentrically. Roughness of the contacting areas can produce similar conditions. This should not influence the readings of the measuring device. Heads with internal measuring devices according to Fig. 5 are to a large extent independent of eccentric pressure.



(*a*) Measuring device.
(*b*) Inner cover.
(*c*) Percussion cap.
(*d*) Ball-shaped force transfer device.

Fig. 5. **Schematic design of a measuring head which is to a large extent insentive to eccentric loading.**

REQUIREMENTS OF THE MEASURING TECHNIQUE.

A high ceiling of frequency is desirable. This is a prerequisite if drop hammers and other forging machines have to be examined. Most other press tools, bending machines and the like, show rather low pressures so that a ceiling of 1000–1500 c/s is sufficient.

An important requirement is the invariability of the calibration curve and insensitivity against harmful influences. Particularly high sensitivity of the heads is not essential, because variations of the properties and sizes of the materials to be deformed are usually large compared with the inaccuracy of the measuring head. A sensitivity of 1·5—2 per cent, which can be easily obtained, is sufficient in most cases.

PARALLEL CONNEXION OF MEASURING HEADS.

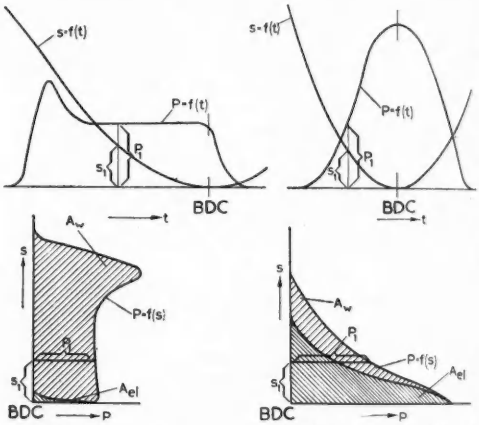
In the case of asymmetrical tools it is frequently necessary to use several heads. Although the individual forces are of interest it is also important to obtain the sum total of all the forces and it is therefore necessary to be able to connect all measuring heads to one integrating meter. This is possible provided that the calibration curves are parallel.

METERS AND INDICATORS.

For practical purposes it is essential to have all the instruments combined in a portable case which should be of moderate weight, that is to say, the case should contain the necessary amplifiers, indicators, oscillographs, power packs and voltage stabilizers. Operation should be simple and foolproof since readings have to be taken in noisy and sometimes inaccessible places. An important point is the voltage stabilizer since large voltage variations in press tool factories are rather the rule.

OSCILLOGRAMS AND THEIR INTERPRETATION.

The curves obtained from the oscillographs are pressure/time curves. Although initially only forces and their variations had to be measured it was soon felt that it would be an advantage to be able to determine the amount of work in kgm. for any particular process. The electrical measuring device can only read pressures and it is necessary to determine also the distance/time curves. From the curves $P = f(t)$ and $s = f(t)$, $P = f(s)$ and therefore $A = \int P\,d\,s$ can be derived. (Figs. 6–7).



Single drawing process of a crank operated press.

Embossing process carried out with a crank operated press.

Figs. 6–7. **Derivation of the pressure/distance diagrams from the pressure/time and the distance/time diagrams.**

Fig. 8 shows the pressure/time relation for a double drawing process. It will be seen that valuable information can be gained from such oscillograms ; among other things they enable the operator to predict whether a particular drawing or pressing process can be carried out on a given press without danger to the bearings and tools themselves.

Fig. 8. **Pressure/time relation for double drawing process.**

CONCLUSIONS.

Practical requirements for electrical measuring devices for press tools have been determined from a large number of experiments. The following points are essential : low weight, independence of eccentric loading, a high degree of invariability of calibration and linear calibration curves. Measuring and indicating devices have to fulfil similar requirements. Measurements in the press tool industry have to be carried out under extremely difficult conditions.

CLASSIFIED ABSTRACTS

Numerous important articles which, for lack of space, cannot be published in full in our main editorial columns, are regularly abstracted in this section. Subscribers may obtain photostat copies of the original articles at cost.

AIRCRAFT

Caproni-Campini Aircraft and Allied Developments in Italy.

C.I.O.S. REPORT, Item No. 5, File No. XII-24.

THIS eighty-page report does not merit careful attention of the designer of aircraft and gas turbines, but is of great interest to the student of technical development and aeronautical history.

It is clear from the report that there was no co-operation between Italy and Germany either in fundamental research or as applied to urgent engineering problems. The Italians' own efforts did not show substantial success, be it in aircraft design or jet propulsion. The Caproni-Campini prototype, well described in the technical press, did not show a thermodynamic propulsive efficiency that would present an advantage over an orthodox airscrew engine combination, whilst the structural complication is extensive. One of the investigations described in the report is of general interest. It goes into the relative merits of installing either exhaust turbo supercharger, or a supercharger driven by a separate reciprocating engine.

Of specific interest to the student of aeronautical history is the documentary evidence revealed by the files of the Italian Air Ministry and General Staff material.

Although of an elementary nature, the calculations of a project submitted to the Italian Air Ministry by Campini, a keen academic engineer, are of interest.

The project comprises two designs :

(1) Twin-jet, single engine Fighter
(2) Twin-jet, twin-engine Bomber

Performance calculations and general lay-outs are given.

BLAST FURNACE PRACTICE

Blast Furnace Operation Under Elevated Top Pressure.

By F. JANECEK. (From *Iron and Steel Engineer, U.S.A.*, Vol. 23, No. 11, February, 1946; pp. 88-94/101, 9 illustrations).

ALTHOUGH blast furnace operation under elevated pressure has to date shown some encouraging results, consistent results have not been obtained because of various delays encountered during the experiment. The lines of the furnace and the changes made to the equipment to make pressure operation possible are as follows : The furnace has a 27 ft. hearth, 30 ft. bosh, and 20 ft. stockline. It is 107 ft. 8½ in. from the iron notch to tunnel ring. The volume is 46,230 cu. ft. from the centre-line of tuyeres to the closed bell. It is equipped with 18 tuyeres and a 15 ft. diameter bell. Thus, the ratio of stockline area to bell area is 1·78, and the ratio of bosh area to hearth area is 1·23, while stockline area is 54·8 per cent of hearth area. The furnace volume above the tuyeres is 80·7 cu. ft. for each sq. ft. of hearth. The bosh angle is 83 deg-17′-32″.

In preparing for the elevated pressure operation, changes above the furnace charging platform consisted of adding two gas lines from the uptakes to introduce the furnace pressure on top of the bell. A third line vented this pressure to the atmosphere (Fig. 1). Valves in these lines are automatically opened and closed by air cylinders, just before the large bell dump. To minimize gas leakage around the stockrod, its diameter was reduced to one inch, and a protecting tube carried to the top of the structure.

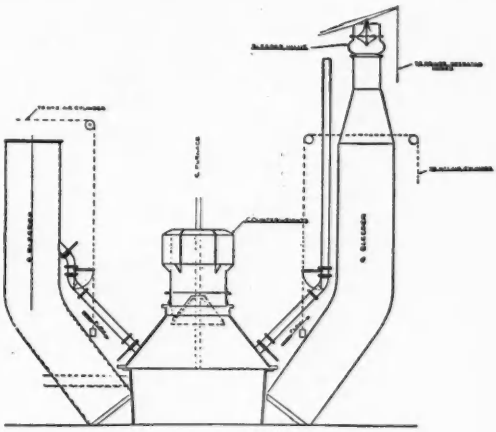


Fig. 1

Special consideration had to be given to getting the water out of the gas washer under the elevated pressure. This was accomplished by introducing a butterfly valve in the water discharge line. This valve is automatically regulated by a float control.

The restricting valve actually controlling and maintaining the 10 lb. pressure in the gas washer, dustcatcher, downcomers and furnace top was constructed of three 30 in. butterfly valves.

The furnace is supplied air by a turbo blower with a capacity of 90,000 cu.ft./min. at 30 lb./sq.in. gauge.

BRAKES AND CLUTCHES

A Cavitation-Free Hydraulic Brake.

By M. Haller. (From *The Brown Boveri Review, Switzerland*, Vol. 32, No. 4, April, 1945, pp. 136-138, 2 illustrations).

THE recently developed hydraulic brake is a great improvement on the existing whirl types. Its main features are its uncommonly wide range of loads and the complete absence of the cavitation phenomenon. The orthodox type of hydraulic brake is regulated by varying the quantity of water. When partially filled, water hammer occurs and cavitation takes place. The new brake remains completely filled with water on all loads ; water can, therefore, be admitted under pressure, which makes efficient cooling possible and prevents water hammer and cavitation. The brake consists chiefly of a pump impeller and the casing. The water flows through the impeller, is reversed in the casing and returns to the impeller. The transmission of power is simply regulated by throttling the water by means of a valve. Another advantage of the new brake is the small moment of inertia of the rotor. The brake has been built up to about 16,000 HP and a maximum speed of about 6,000 r.p.m. It can be built for a single or for both directions of rotation.

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CELLULOSE

The Change of Cellulose in the Viscose Process.

By Z. CSUROS and L. MOLNAR. (From *Technika, Hungary*, Vol. 1946, No. 248, January, 1946, pp. 1-9, 4 illustrations).

THE authors investigated the change of the polymerization degree in cellulose during the manufacturing process in the Hungarian Viscose Works in Nyergesujfalu. Investigations have been carried out with both the Swedish Super Forshaga and the German Kellner-Partington cellulose. After two days previous maturing, the polymerization degree of both types has decreased from 620 to about 220 deg. and this remained unchanged in course of further processing. This fact has led to the conclusion that the great changes in viscosity are independent from the changes in the polymerization degree.

ELECTRIC MACHINERY

Enhanced Transformer Protection through Inrush-proof Ratio Differential Relays.

By G. BERTOLE. (From *The Brown Boveri Review, Switzerland*, Vol. 32, No. 4, April, 1945, pp. 129-133, 7 illustrations).

AFTER some general information on the differential protection of transformers as used until now, the influence of inrush phenomena on these protecting devices is explained. The means are then exposed by which it was tried to avoid the influence of inrush currents and especially the undesired tripping of differential relays. Though solutions were found to make the relays selective to inrush currents and fault currents, these deprive the differential relay of its simplicity of design. Recently, a new inrush-proof ratio differential relay, the protecting value of which remains unchanged during the transformer switching-in period was developed. It is based on the phenomenon that the half-waves on the side of the zero line of an inrush-current are throughout of much lower amplitude than the half-waves on the other side. Such a current can be considered as a current pulsating with a frequency of half the nominal value, especially as regards its effect on the torque of a relay. Thus, the standard ratio differential relay was modified so that its moving system vibrates under torque impulses of half normal frequency. For normal inrush phenomena these vibrations continue as long as the current impulses and prevent the relay contacts remaining closed long enough to enable the tripping operation to be completed. Should, however, a short circuit take place during normal operation or switching of the transformer, the relay contacts would quickly close under the prevailing effect of the normal frequency, and the circuit breakers would be tripped with the small time lag inherent in the built-in small contactor. The relay operates correctly for frequencies between $\pm$ 15 per cent. of the nominal.

ELECTRIC MEASUREMENTS.

Measurement of Small Electric Values.

By J. BEDNARIK. (From *Slaboproudy Obzor, Czechoslovakia*, Vol. 7, No. 12, Jan., 1946, pp. 4-8, 10 illustrations).

POSITIVE feedback-type amplifiers for accurate measurements of small low frequency voltages and currents are described and results of comparative measurements of accuracy of these amplifiers with the accuracy of the negative feedback amplifiers are given in this article. The positive feedback amplifiers possess the same properties as the instrument transformers, used for measuring high voltages or big currents and the voltage or current produced is independent of the impedance connected to the terminals of the output circuit (i.e. of the kind of applied measuring apparatus). This condition cannot be fulfilled satisfactorily with negative feedback amplifiers.

JET PROPULSION

Facts Behind Jet-Engine Performance.

By R. P. KROON, Manager of Engineering Aviation Gas Turbine Div., Westinghouse Electric Corp. (From *Machine Design*, U.S.A., Vol. 18, No. 1, Jan., 1946, p. 132, 1 illustration.)

RECENT models of the first all-American designed jet engine—the Westinghouse 19B—weigh less than half a pound per pound of thrust. Like other jet engines, they may be regarded as essentially machines for increasing the speed of a mass of air, in this case some 50 tons per hour. By imparting to this amount of air a jet velocity of 1200 m.p.h. they develop a thrust of 1400 lb.

In operation, air from the atmosphere enters the circular throat of the engine at the air speed of the plane, passing through a stationary ring of air-foil-shaped aluminium blades which increase its speed to about 600 m.p.h. before entering the compressor. The 6-stage axial-flow compressor has a forged aluminium rotor approximately 1½ ft. long, provided with nearly 200 aerofoil blades and rotating at 18,000 r.p.m. Air is discharged from the compressor into the combustion chamber where it is mixed with injected fuel and ignited. The resultant rate of heat release per cubic foot per hour is a thousand times as great as that in a central station boiler, the air particles spending only 1/100-sec in the combustion chamber.

Combustion products leave the chamber at 1500 deg. F. and pass through the blades of the single-stage turbine which drives the compressor. Of the total energy in the gases the major portion—about 2400 h.p.—is absorbed in the turbine, from which the gases leave at 600 m.p.h. and a pressure somewhat above that of the atmosphere. Further expansion in the 2-ft. long jet nozzle increases the velocity to 1200 m.p.h., the actual speed being controlled by an adjustable tailpiece. The air particles acquire this velocity change in only 1/700-sec., which corresponds to an acceleration of nearly 20,000 g. At sea level the thrust is about 1400 lb. regardless of plane speeds, hence the thrust horse-power is directly proportional to plane speed, being equal to 1400 horsepower at 375 m.p.h.

A smaller engine of similar design—the 9·5A—also has been developed. Originally intended for an American "buzzless" robot bomb the unit holds promise, in a modified form, as a gas turbine drive for light planes, helicopters, cabin supercharger, and electric generators. With a top speed of 34,000 r.p.m. the 9·5A can deliver a thrust of 275 lb., yet is only 9·5 inches in diameter.

MATHEMATICS

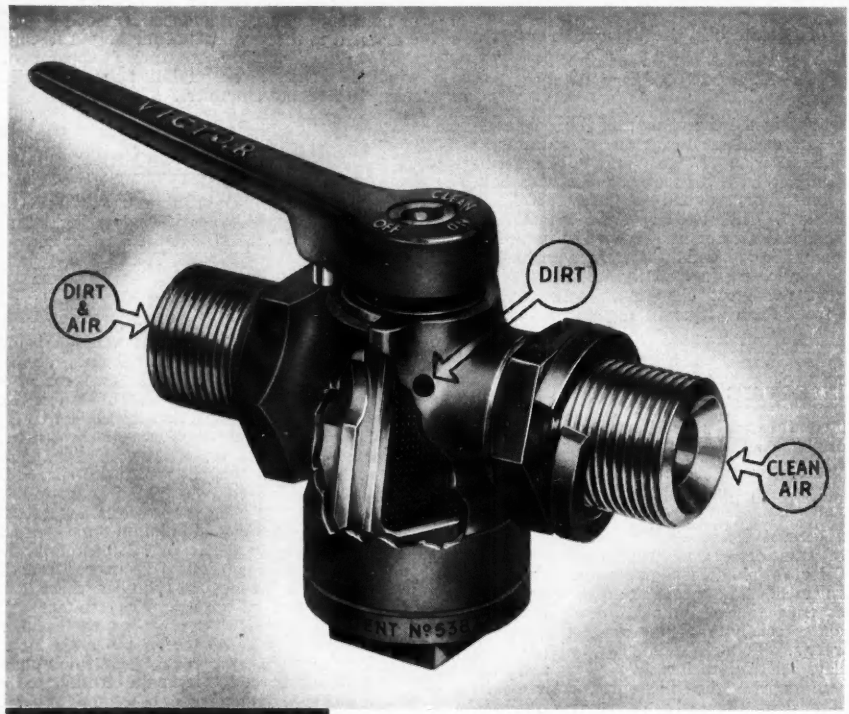
Graphic Integration as a Supplement to the Method of Conform Representation.

By J. HAK. (From *Elektrotechnicky Obzor, Czechoslovakia*, Vol. 34, No. 17-18, December, 1945, pp. 245-254, 22 illustrations).

THE method of conform representation is used to determine the distribution of potential in cases where the gap is bordered by sectional straight lines. Its practical application is limited to cases where the complete integration of the basic equation

$$z = k \int (t-\tau_1)^{-\alpha_1/\pi} (t-\tau_2)^{-\alpha_2/\pi} \dots (t-\tau_n)^{-\alpha_n/\pi} dt + C$$

which assigns to any point of the main plane t one definite



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point of the auxiliary plane z, is possible; in this equation, k and C are two usually complex constants specifying the proper situation of the z-plane with respect to the t-plane and fixing the scale in the z plane. $\tau_1, \tau_2, \ldots \tau_n$ correspond to the vertices of the polygon and $\alpha_1, \alpha_2, \ldots \alpha_n$ are the supplementary angles of the polygon at these vertices. This often leads to elliptic integrals and functions for even the relatively simpler cases and to hyperelliptic and unsolvable integrals in many important cases. As, however, the solution of this equation is indispensable, this method has serious limitations. The article shows how in such cases a graphic method can be used to integrate the basic equation. The integration itself is carried out in two stages and in the vicinity of points for which the value of the function becomes infinite, the theorem of the mean value of integrals is applied. An example is given showing both the conventional and the graphic methods; another more complicated example shows the application of the graphic method in a case where no solution could be obtained by the conventional methods.

PLASTICS

Cold Moulded Plastics.

By B. F. HANTZ, Chief Engineer, American Insulator Corp. (From *Plastics, U.S.A.*, Vol. 4, No. 1, January, 1946, pp. 46, 50-52, 131, 9 illustrations.)

COLD moulding differs from the customary moulding methods in that the parts are not completed when they come from the mould but must be baked in an oven. There they take their final form as the result of the oxidation and polymerization of the oils and binder in the compound. Because of these differences, cold moulding must be considered as an entirely separate class of moulding.

Cold moulding materials are divided into three groups, depending upon their binders: (1) organic binder—bituminous—asphalt; (2) organic binder—synthetic resin—phenolic; and (3) inorganic binder—cement.

The organic type of cold moulding material will withstand a higher operating temperature than most other moulding powders. It is also more resistant to an electric arc, making it very suitable for electric switches and fuse plugs.

Certain classes of inorganic cold moulded materials actually improve with age. They will stand an operating temperature of 1000 deg. F.

The parts are usually produced on hydraulic rather than mechanical pressures because it is necessary to have sufficient pressure in unlimited amounts available at all times. The cold moulding presses range in capacity from 5 to 200 tons, which is sufficient to produce most parts required by the trade.

The moulding cycle is very rapid. An experienced operator can produce 500 to 700 pieces per hour from a single-cavity mould. This production can be increased if a larger number of cavities are used in the mould. After the parts are pressed to shape, they are packed on metal trays with perforated bottoms. These allow a free circulation of air, essential for most efficient baking. They remain on the trays at room temperature for 24 to 72 hours. During this period, volatile plasticizers evaporate. The trays are then put on trucks which hold from 20 to 40 trays, and the trucks are pushed into the oven. The baking period varies from 8 hr. to 48 hr., depending upon the size and design of the piece.

REFRACTORY MATERIALS

Refractories in Turbine Blades.

C.I.O.S. REPORT, Item No. 1, 18, 21 & 25, No. XXXI-22.

THIS report deals with the application of special high temperature refractories to gas turbine nozzles and blades and also to their use in heat exchangers for gas turbines.

As the thermal efficiency of gas turbines depends on the magnitude of the gas inlet temperature, and even with good alloys the operating temperature is only in the region of 1200 deg. F, the Germans, lacking the elements of high temperature alloys, were compelled to investigate the possibility of utilizing non-metals in turbine construction.

Several German firms turned their attention to this work, and the report describes the partial success they attained. The issue is still undecided, yet to many of those who have seen these attempts among the exhibits at the Science Museum, the idea might appeal, and the report appears to be worth studying.

Since ceramics are brittle, attention of the Germans was divided between compositions for high strength and thermal shock resistant compositions.

For strength, sintered alumina score highest with a tensile strength of 38,000 pounds per square inch at room temperature. This value decreases very little up to 1830 deg. F. At 2200 deg. F. the tensile strength is still 20,000 pounds per square inch. The specific gravity of ceramics is roughly one half of that of metals. Thus, in a centrifugal field of 10,000 gravity, a three inch high blade would be stressed to 4,300 pounds per square inch at the blade root.

As to the thermal-shock resistant blades, these were of compositions containing silicon carbide, "Ardostan" and others. The coefficient of thermal expansion for Ardostan is $1{\cdot}1 \times 10^{-6}$ deg. C, only about a sixth of that of sintered alumina. It makes Ardostan very insensitive to sudden changes of temperature. Tensile strength is around 6,500 pounds per square inch with no temperature dependence up to 1830 deg. F.

The report describes a heat exchanger very much on the lines of the Ljungstrom air-pre-heater known from steam raising plant practice. The rotating part of this heat transfer device consists of a ceramic drum, the weight of which, for an aircraft turbine, is estimated to be 0·1 to 0·15 kg. per horse power.

SCIENTIFIC INSTRUMENTS AND PRECISION TOOLS

The Rotascope.

(From *General Electric Review, U.S.A.*, Vol. 49, No. 1, Jan., 1946, p. 50).

A new optical device, the rotascope, was developed to rotate a beam of light to produce a stationary image of a rotating object. Originally designed to permit continuous observation of the tip of a rapidly rotating blade, it will have many additional applications where it is desired to continuously view rotating parts.

Rotascope viewing differs from stroboscopic in many respects. The latter is intermittent, depending on a rapidly operating shutter or a flashing light source, with persistence of vision giving an impression of a stationary object. Rotascopic viewing, however, gives a continuous rather than intermittent view of the rotating object; can be used at very low speeds, and can be used to show transient, intermittent, or random occurrences of the rotating part.

A prism with one silvered surface is interposed in the line of sight and then rotated about this line. As the

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prism is rotated, the image seen through it rotates, but at twice the angular velocity of the prism. A rotating object viewed through this rotating prism thus seems to stand still, a " mirror image " being produced. An auxiliary mirror or optical system can be used to rectify the image. For viewing more rapidly rotating objects, two or more rotascope prisms are used in series, rotating in opposite directions.

WELDING

Centralized Supply of Welding Current for A. C. Welding.

By M. G. WOLFF. (From *Bul. Soc. Francaise des Electriciens, France*, Ser. 6, Vol. 6, No. 55, January, 1946, p. 23-27, 10 illustrations).

A CONTINUOUS arc between metallic electrodes is difficult to maintain in the case of alternating current, and coated electrodes must therefore be used for welding. But even so, care must be taken in a.c. welding to maintain a shorter arc than is permissible in d.c. welding. While the same type of weld is produced by either type of current, the properties of the material in a weld produced by a.c. current are slightly better, and this refers in particular to the ductility of the deposited metal. This is explained by the fact that with the shorter arc employed in a.c. welding, less atmospheric oxygen and nitrogen are admitted to the weld metal.

The greater steadiness of the a.c. arc is due to the fact that its magnetic field is damped by hysteresis and by Foucault currents. The difficulties encountered with maintaining a stationary arc in d.c. welding are therefore absent in a.c. welding even when using large diameter a.c. welding electrodes and large current intensities.

In planning the distribution system of a centralized a.c. welding plant, reactance of the cable connections to the individual welding stations must also be taken into account. But it is interesting to note that even so arc stability is better ensured in a.c. distributing systems than in d.c. distributing systems. From a comparison of a d.c. welding plant with an a.c. plant it was found that the power consumption per kg. weld metal deposited amounted to 7·5 KW. hrs. in the d.c. plant and 3·3 KW hrs. in the a.c. plant. On the basis of equal current output, the cost of the centralized d.c. plant is two to three times that of the a.c. plant, and its weight is 65 per cent greater. French manufacturers of welding electrodes are now investigating the characteristics of a.c. electrodes for a normal voltage of 50 V.

Modern Processes for Automatic Control of the Welding Cycle of Spot and Seam Welding Machines.

By A. DUTOIT and W. HEINZ. (From *The Brown Boveri Review, Switzerland*, Vol. 32, No. 4, April 1945, pp. 142-147, 9 illustrations).

MECHANICAL switches, the simplest and least expensive device for interrupting and switching on the welding current, are still used for most resistance welding operations. Where high power welding machines for special work with frequencies over 50 cycles, or new manufactures for which a special welding technique must first be developed, are concerned, static methods of control can render valuable service. In particular, this can be the case with seam welding machines. In the case of a machine where the alternating current is switched on and off at no definite point on the current curve (asynchronous switches), very high current peaks occur which have adverse effects on the supply system as well as on the switch contacts. These are avoided by the use of switches controlled as a definite function of the system frequency (synchronous switches). For this purpose a mutator switch consisting of two grid-controlled single-anode mutators or one two-anode mutator with grid control, was developed which is connected in series with the welding transformer and which operates exclusively as an electrical switch so that the circuit is always opened or closed at the maximum value of the alternating voltage. Welding times and intervals can be adjusted on the equipment as a function of half-cycles. A small cathode-ray oscillograph is generally employed to check the mode of operation of such static switches. The new switch has produced sound work where absolute switching time accuracy and extremely short welding times are called for, and which cannot be realized with even the most modern mechanical switches.

WOOD AND FOREST PRODUCTS

Visits to Targets Connected with the German Plywood Industry.

B.I.O.S. REPORT, No. 41 (final report), Item No. 22.

THE report describes the products, the methods of production and testing, and the wood adhesives common in the German plywood industry. Fourteen visits were made, and each is described separately. Some of the more important ones will be dealt with in the present abstract.

The Traxel Firma Works, nr. Frankfurt, are engaged on plywood manufacture. Most of the machinery used is of the conventional type. Before the war, up to 1940, gaboon and Oregon pine were used, but since 1940, only beech has been available. Hide glue was used formerly, by Kaurit has superseded this. Up to 1940, the glues for plywood were mainly blood, casein or blood/casein mixes of conventional type without toxic additions. Shortage of raw casein and dried blood forced the use of Kaurit exclusively, usually extended with 10 per cent to 40 per cent of beech wood-flour or potato meal plus 5 per cent to 10 per cent of water. The hardener was of the hot type. No Tego was used. This state of affairs is typical for the war-time plywood industry in Germany.

Metal-faced plywood is now made with a glue called Leukolat. It may be of a rubber type, and is brush-applied, allowed to dry, pressed at 130-140 deg. C. and cooled in the press.

The Chemische Werke Albert at Wiesbaden is a plastics firm, whose principal output was urea-formaldehyde-butanol varnishes. In 1942, owing to the failure of urea-formaldehyde resin glued wooden aircraft in North-Africa, the firm developed a cold-setting phenolic glue. Such glues were used in Russian aircraft. In comparison with the development of phenolic adhesives in Great Britain, the progress in Germany has been extremely small.

Two visits to scientific workers are of particular interest. One was to Prof. Edgar Mörath, nr. Salzburg, the wood technologist, the other to Prof. Vieweg, Darmstadt. The former is a specialist in plywood, the latter a physicist who has specialized on the determination of the physical constants of plastics. Prof. Vieweg was also connected with research on foamed plastics, particularly foamed polystal. As to the glue polystal made by I. G. it was said to be a di-iso-cyanate, and imparted great flexibility to the plywood. It is considered the most important adhesive produced in Germany during recent years ; since 1942 in large scale production.

One of the most interesting firms visited during the tour was that of Erwin Behr, Wendlingen, nr. Stuttgart. The firm specializes in moulded plywood products, the execution of which is of a very high order. Great technical skill is shown in the tooling, and the use of laminated wood is clever. The technique of manufacture of aircraft components is worth studying.

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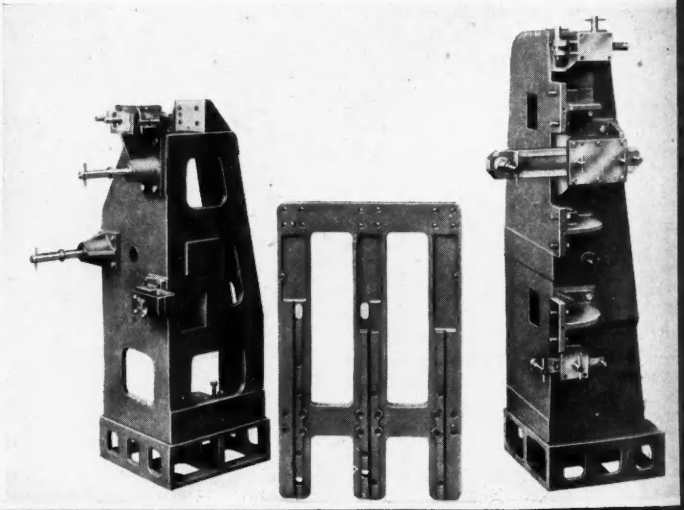

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